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Predmore et al.

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[54] **COOLING JOINT CONNECTION FOR ABUTTING SEGMENTS IN A GAS TURBINE ENGINE**

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[21] Appl. No.: **719,666**

[22] Filed: **Sep. 25, 1996**

[51] **Int. Cl.**[6] **F01D 25/26**

[52] **U.S. Cl.** **415/134; 415/139**

[58] **Field of Search** 415/134, 135, 415/136, 137, 138, 139

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,863,634	12/1958	Chamberlin et al.	415/134
4,537,024	8/1985	Grosjean	415/139
4,650,394	3/1987	Weidner	415/115
4,676,715	6/1987	Imbault et al.	415/116
5,158,430	10/1992	Dixon et al.	415/134
5,167,485	12/1992	Starkweather .	
5,374,161	12/1994	Kelch et al.	415/139
5,609,469	3/1997	Worley et al.	415/115
5,634,766	6/1997	Cunha et al.	415/115

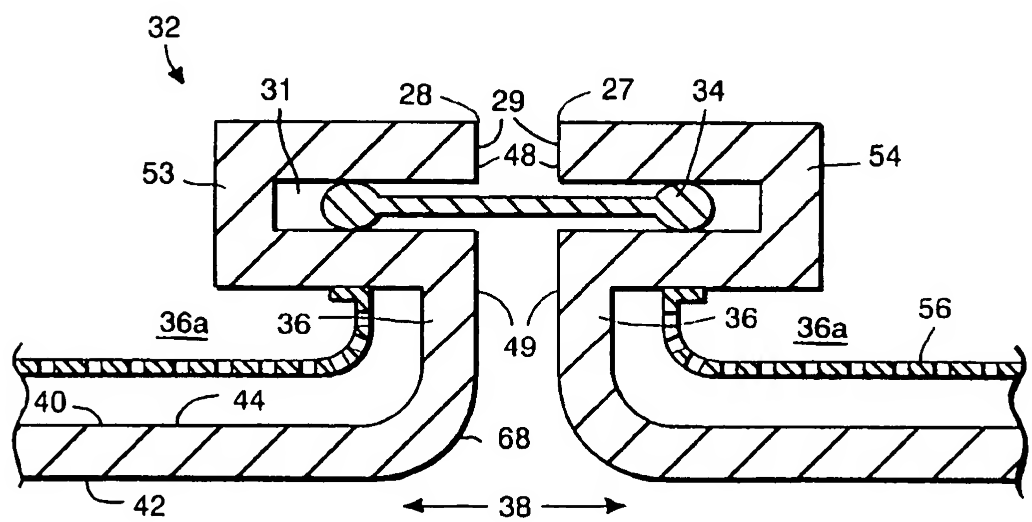
Primary Examiner—John T. Kwon

Attorney, Agent, or Firm—Nixon & Vanderhye

[57] **ABSTRACT**

A seal joint including a sealing member for sealing segmented components of a gas turbine engine is spaced from the gas path to provide more effective cooling. The segments include radial extension flanges spacing the seal joint from the gas path. Various cooling arrangements are adaptable to the modified segments to provide for open or closed circuit impingement or convection cooling plus film cooling. Excessive thermal gradients are avoided as the entire corner of the segment is evenly cooled. Moreover, air usage at the joint is bounded by seal leakage with no extra dedicated cooling air.

17 Claims, 2 Drawing Sheets



1, 8, 9

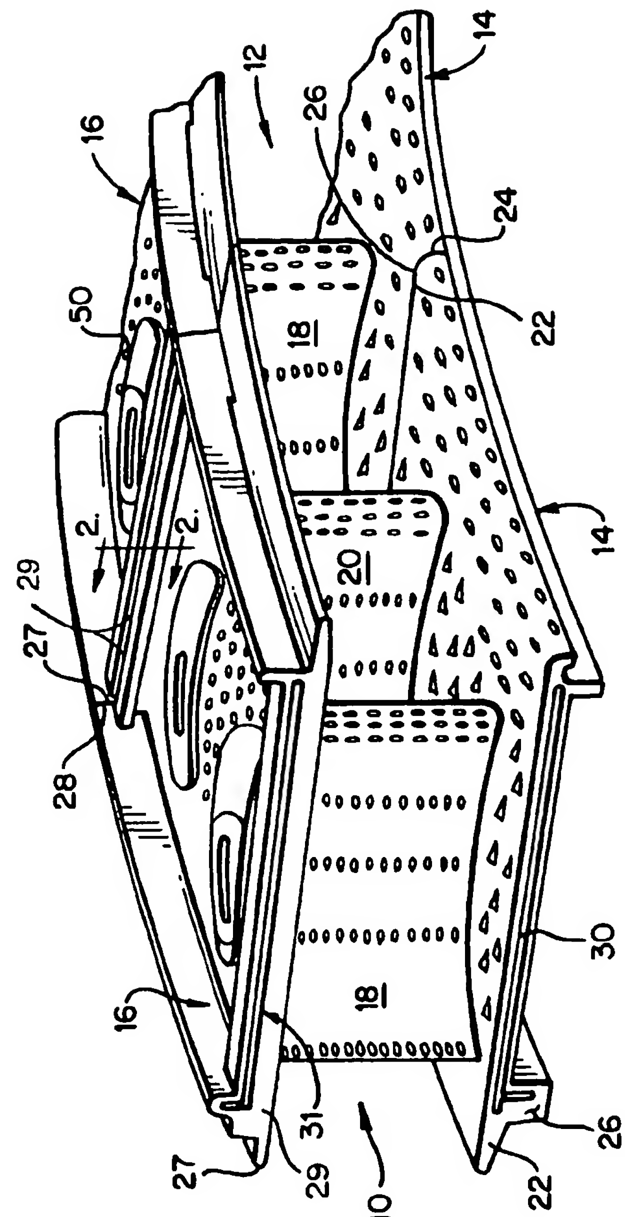


FIG. 1

1, 8, 9

Vane — vane

FIG. 2

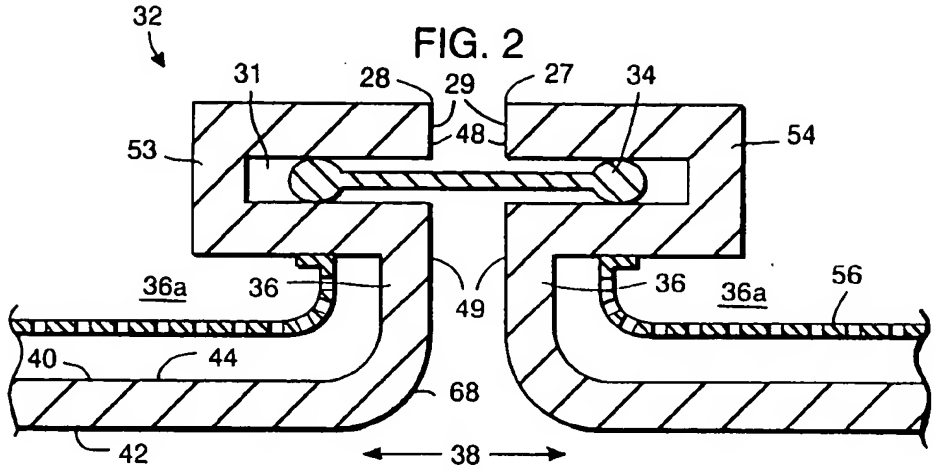


FIG. 3

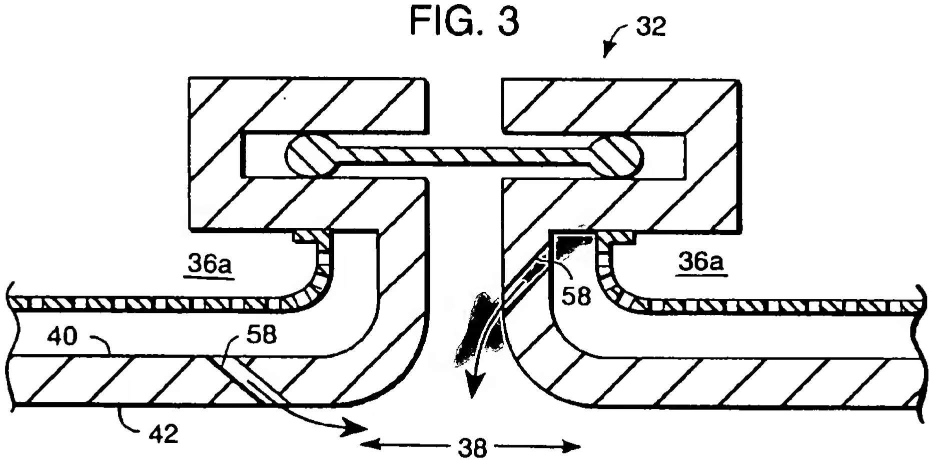
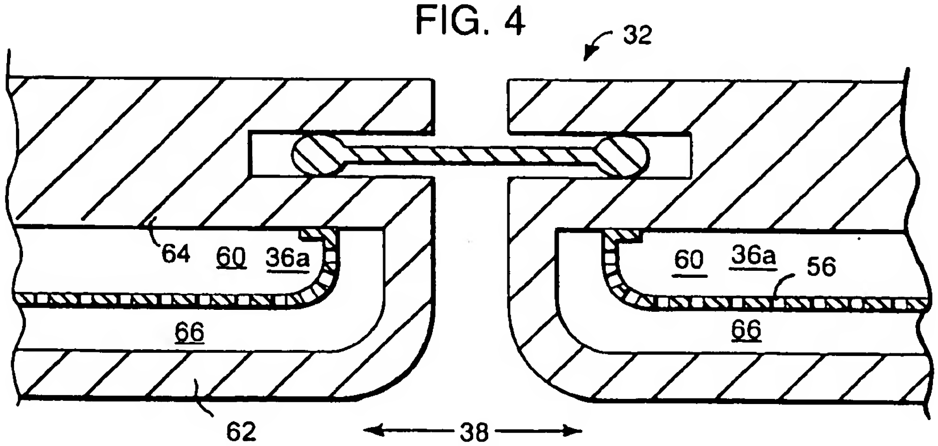


FIG. 4



COOLING JOINT CONNECTION FOR ABUTTING SEGMENTS IN A GAS TURBINE ENGINE

TECHNICAL FIELD

This invention relates to gas turbine engines and, more particularly, to an improved joint connection for the abutting edges of circumferentially extending segments in gas turbine engines such as nozzles, buckets and shrouds.

BACKGROUND

An important consideration in the design of gas turbine engines is to ensure that various components of the engine are maintained at safe operating temperatures. This is particularly true for elements of the combustor and turbine, which are exposed to the highest operating temperatures in the engine, and which include turbine nozzles, buckets and shrouds. The purpose of the turbine nozzle is to direct hot gas at an optimal angle, and within an annular gas path, to cause the adjacent following bucket row to rotate, producing power. The purpose of the shroud is to define the gas path radially outward of the rotating bucket row.

In the turbine section of gas turbine engines, high thermal efficiency is dependent upon high turbine entry temperatures. These entry temperatures, in turn, are limited by the heat which the materials forming the turbine nozzles, buckets and shrouds can safely withstand. In cases in which the gas path temperatures are above the material limitations, the gas path surfaces of these components must be cooled to survive. Thus, in addition to improvements in the types of materials and coatings used to fabricate these components, continuous air cooling has been employed to permit the environmental operating temperature of the turbine to exceed the melting point of the materials forming the components without affecting their integrity.

A number of air cooling techniques have been used in an attempt to effectively and uniformly cool the components of the turbine, combustor and other portions of gas turbine engines. The turbine nozzle segments, for example, are conventionally cooled by a combination of air impingement, film, pin fins and convection/film holes. Each nozzle segment, which comprises inner and outer bands interconnected by fixed nozzle guide vanes, is the beneficiary of a combination of such cooling methods to reduce both the internal and external temperature of the nozzle bands and guide vanes.

One problem area in the cooling of turbine nozzle segments is at the joint connections between adjacent nozzle segments. In order to prevent thermal hoop stresses and to facilitate fabrication, the inner and outer bands supporting the nozzle guide vanes must be segmented, i.e., the nozzles are formed by a number of arcuate segments, each having arcuate-shaped inner and outer bands arranged to extend circumferentially about the turbine case, and abut one another at their side edges. Conventionally, a slot or pocket is formed in each side edge of adjacent turbine nozzle segments and a sealing member is located within and between the cooperating slots of adjacent segments to create a seal therebetween. It has been found, however, that the sealing area between adjacent segments is cooled less effectively than the remainder of the inner and outer bands of the nozzle segments, which creates an uneven heat distribution along the nozzle segments.

Attempts have been made to improve cooling of the joint connection or seal area between abutting turbine nozzle segments, but problems have been encountered with each

design. In one design, pressurized air is introduced on the non-gas path side of the sealing member and the resulting leakage to the gas path is used to cool the material. In this design, however, leakage can be unevenly distributed, creating local hot spots, and may lead to "leaky" designs.

Another technique is disclosed in commonly owned U.S. Pat. No. 5,167,485, wherein a number of channels or grooves are formed in the slot of the joint connection forming an airflow path in the seal region such that cooling air is permitted to flow onto one side of the sealing member, into each of the slots in the abutting inner and outer bands of the nozzle segments, around the edges of the sealing member and into the channels or grooves in the inner or outer wall of the slots to the opposite side of the sealing member. Slot cooling, however, requires additional dedicated coolant and results in reduced efficiency.

In another design, the corner adjacent the edge region is overcooled such that the material is cooled by conduction. Conduction cooling, however, can create unacceptable lateral thermal gradients and alone may be insufficient. In yet another method, the material is cooled by film from an adjacent region. Film cooling, however requires additional dedicated coolant and adds manufacturing expense. Air used to cool nozzle, bucket or shroud segment edges by leakage, slots or film bypasses the combustor, thereby increasing emissions and reducing turbine performance. In general, it is advantageous to minimize air usage.

DISCLOSURE OF THE INVENTION

It is, therefore, among the objectives of this invention to provide joint connections between circumferentially arranged segments of turbine (or other rotary machine) components such as nozzles, buckets and shrouds, which effectively cool the seal regions between adjacent segments. In particular, the seal joint having the sealing member therein in accordance with the present invention is located farther away from the gas path than in previous arrangements, creating more room to actively cool the edge. Moreover, excessive thermal gradients are avoided as the entire edge corner is evenly cooled. Air usage at the joint is bounded by seal leakage with no extra dedicated cooling air required.

This and other objects of the invention are achieved by providing, for example, a plurality of circumferentially adjacent nozzle segments, each having a warm side surface and a cool side surface. Each segment includes an inner band, an outer band, and one or more guide vanes extending therebetween. The inner and outer bands each have a pair of side edges or faces which are adapted to substantially abut like edges or faces of adjacent segments.

Each side face of each band is formed with an elongated slot extending axially along the side face, the slot opening toward a corresponding opposing slot in an adjacent face of an adjacent segment. Sealing members are employed between the opposed slots of adjacent segments with a gas path on one side of the sealing member and a non-gas path on the other side.

In exemplary embodiments of this invention, the sealing members in the inner and outer bands are located farther away from the gas path than in previous designs.

In a first exemplary embodiment, a seal joint is defined at a radially outermost end of adjacent radial extension flanges. The radial extension flanges extend outwardly from the adjacent segments radially away from the gas path. Cooling air flows through an impingement plate to cool the sealed joint. In a second exemplary embodiment, film slots are

provided in the segments to provide conventional film cooling along with open-circuit impingement or convection cooling. In a third exemplary embodiment, a closed circuit region is provided for closed-circuit impingement or convection cooling. In each configuration, the seal joint is radially spaced from the gas path by an amount substantially corresponding to the length of the radial extension flanges.

In its broader aspects, therefore, the present invention thus relates to a segment for a circumferential component of a rotary machine. The segment includes a longitudinal section extending in a first direction, a seal joint section including a side edge face extending in a second direction substantially perpendicular to the first direction, and a slot formed in the side edge face extending from the face in the first direction, dividing the face into a radially inner face portion and a radially outer face portion. The slot is shaped to receive a sealing member in cooperation with an adjacent slot of the adjacent segment, wherein the radially inner face portion is longer than the radially outer face portion.

In this context, the invention also relates to a gas turbine engine including at least one section having a plurality of the circumferentially adjacent segments.

In another aspect, the invention relates to a segmented component of a turbine engine including a plurality of pairs of adjacent segments, each pair of adjacent segments including a seal joint having a slot therein shaped to receive a seal member. The seal joint is radially spaced from a gas path by an amount sufficient to enable active cooling of the seal joint.

In still another aspect, the invention relates to a method of sealing and cooling circumferentially adjacent segments in a gas turbine engine. The method includes the steps of locating a seal joint slots in adjacent segments, radially spaced from a gas path by an amount sufficient to enable active cooling of the seal joint slots, and inserting a sealing member in the seal joint slots of adjacent segments to seal the adjacent segments.

BRIEF DESCRIPTION OF THE DRAWINGS

The structure, operation and advantages of the presently preferred embodiments of this invention will become further apparent upon consideration of the following description, taken in conjunction with the accompanying drawings, wherein:

FIG. 1 is a schematic, perspective view of two abutting turbine nozzle segments of a gas turbine engine employing the side edge seal of this invention;

FIG. 2 is a cross sectional view of the abutting nozzle segments taken generally along line 2—2 of FIG. 1

FIG. 3 is a cross sectional view of alternative abutting nozzle segments taken generally along line 2—2 of FIG. 1; and

FIG. 4 is a cross sectional view of further alternative abutting nozzle segments taken generally along line 2—2 of FIG. 1.

BEST MODE FOR CARRYING OUT THE INVENTION

Referring now to FIG. 1, a first turbine nozzle segment 10 and part of a second nozzle segment 12 are shown abutting each other, forming a portion of an essentially continuous, circumferentially extending nozzle stage within the turbine section of a gas turbine engine. For purposes of the present disclosure, only the construction of turbine nozzle segment 10 is discussed in detail, it being understood that the other nozzle segment 12, and all other nozzle segments within the nozzle assembly are structurally and functionally identical. In addition, it is to be understood that the invention here is equally applicable to the construction of annularly segmented turbine buckets and shrouds.

The turbine nozzle segment 10 comprises an inner band 14, an outer band 16 and a pair of nozzle guide vanes 18, 20 connected between the inner and outer bands 14, 16. While a two vane segment is depicted, it is recognized that the segment can have one or any other number of vanes. The inner band 14, outer band 16, and nozzle guide vanes 18 and 20 are shown as including film cooling holes 50 which serve as passages to provide cooling air through the parts for convection cooling and to surfaces exposed to hot gases for film cooling. The inner band 14 of nozzle segment 10 is formed with opposite side edges 22, 24, each having an edge face 26. Similarly, the outer band 16 of nozzle segment 10 is formed with opposite side edges 27, 28 each having an edge face 29. In the assembled position, the side edges 22, 24 of the inner band 14 and the side edges 27, 28 of the outer band 16 of adjacent nozzle segments substantially abut to form an essentially continuous, annular nozzle assembly.

The side edges 22, 24 of the inner band 14 and the side edges 27, 28 of the outer band 16 are each formed with a longitudinally extending pocket or slot 30, 31, respectively. For purposes of the present disclosure, the slot 31 in abutting side edges 27, 28 of the outer bands 16 of segments 10, 12 is described in detail, it being understood that the slots 30 in the inner bands 14 thereof are generally similar in structure and function.

Referring now specifically to FIG. 2, the joint connection between the outer bands 16 of the nozzle segments 10 and 12 is illustrated, wherein the side edge 28 of the outer band 16 of segment 10 substantially abuts the side edge 27 of the outer band 16 of segment 12. A slight leakage gap between the abutting outer bands 16 is exaggerated in FIGS. 2–4 for purposes of illustration only.

The joint connection according to the present invention is embodied in a seal joint 32 including opposed slots 31, which receive a sealing member 34. The seal joint 32 is disposed at distal ends of adjacent radial extension flanges 36, which extend substantially perpendicular to a turbine gas path 38. By virtue of the radial extension flanges 36, the seal joint 32 is spaced away from the gas path 38 by an amount sufficient to enable active cooling of the seal joint 32. That is, the seal joint 32 is spaced from the gas path 38 such that coolant can be directly contacted with the radial extension flanges 36 and the seal joint 32 as opposed to cooling by conduction or the like.

The segmented components each include a longitudinal section 40, which extends substantially parallel to the gas path 38 and is continuous with the radial extension flanges 36 as shown in FIG. 2.

The face 29 of the side edges 27 and 28 is divided by the slot 31 and sealing member 34 into a radially outer face portion 48 and a radially inner face portion 49. By virtue of the radial extension flanges 36, the radially inner face portion 49 is longer than the radially outer face portion 48.

The slot 31 is formed in the adjacent components by one of the adjacent components having a C-shaped section 53, and the other of the components having a reverse C-shaped section 54. The C-shaped section 53 disposed adjacent the reverse C-shaped section 54 delimits the slot 31 receiving the sealing member 34. Sections 53 and 54 are located at distal ends of the flanges 36, and extend substantially parallel to the gas path, such that the slot 31 and sealing

member **34** also extend substantially parallel to the gas path. As such, an undercut area **36a** is defined between the seal joint **32** and the longitudinal section **40**.

An impingement plate **56** is mounted to each segment of the seal joint **32** and extends adjacent and substantially parallel to the radial extension flanges **36** and then substantially parallel to the gas path **38** adjacent the longitudinal section **40**. Cooling air is flowed through the impingement plates **56** in a conventional manner, providing open-circuit impingement or convection cooling along the longitudinal section **40**, the radial extension flanges **36** and the seal joint **32**.

As shown in FIG. **3**, in an alternative arrangement, film slots **58** are provided in the longitudinal section **40** to additionally provide conventional film cooling along with the open-circuit impingement or convection cooling. The joint arrangement is otherwise similar to that shown in FIG. **2**.

FIG. **4** illustrates yet another alternative arrangement including a closed annular circuit region **60** providing closed-circuit impingement or convection cooling. The region **60** is defined by an impingement plate **56** and an outer wall portion **64** of the segment. Coolant, such as steam, is directed from a steam circuit into the closed annular circuit region **60** via a pipe. After impingement, the coolant is returned to the steam circuit via path **66** defined between impingement plate **56** and an inner wall portion **62**. Steam can be used as the coolant in this configuration because the cooling region is closed from the gas path **38**.

As noted above, by virtue of the radial extension flanges **36**, the seal joint **32** is spaced from the gas path **38** by an amount sufficient to enable active cooling of the seal joint **32**, which is substantially farther away from the gas path than in prior art configurations. Further, the rounded corners **68** tend to reduce conduction of heat through the metal from the hot gas surface **42** to the cold metal surface **44**. As a result, excessive thermal gradients are avoided as the entire corner is evenly cooled. Air usage at the joint is bounded by seal leakage with no extra dedicated coolant required.

While the invention has been described in connection with what is presently considered to be the most practical and preferred embodiments, it is to be understood that the invention is not to be limited to the disclosed embodiments, but on the contrary, is intended to cover various modifications and equivalent arrangements included within the spirit and scope of the appended claims.

What is claimed is:

1. In a gas turbine engine including at least one section comprising a plurality of circumferentially adjacent segments, each of said segments comprising:

a longitudinal section extending in a first direction;

a radial extension flange continuous with said longitudinal section and extending in a second direction substantially perpendicular to said first direction; and

a seal joint section continuous with said radial extension flange and defining a slot, said slot being shaped to receive a sealing member in cooperation with an adjacent slot of an adjacent segment, wherein said seal joint section is radially spaced from said longitudinal section by an amount substantially corresponding to said radial extension flange defining an undercut area between said longitudinal section and said seal joint section.

2. The segment of claim **1**, further comprising an impingement plate secured adjacent said longitudinal section, a portion of said impingement plate extending substantially parallel to said radial extension flange.

3. The segment of claim **2**, further comprising at least one film slot extending through said longitudinal section.

4. The segment of claim **2**, further comprising a closed circuit cooling region defined in part by said impingement plate.

5. The segment of claim **2**, wherein said impingement plate comprises a plurality of coolant flow apertures therethrough.

6. A segmented component of a turbine engine including a plurality of pairs of adjacent segments, each pair of adjacent segments comprising a seal joint having a slot therein shaped to receive a seal member, said seal joint being radially spaced from a gas path defining an undercut area between said seal joint and said gas path sufficient to enable active cooling of said seal joint and said undercut area.

7. The segmented component of claim **6**, wherein one of said pair of adjacent segments comprises a substantially C-shaped section and the other comprises a substantially reverse C-shaped section, said C-shaped section and said reverse C-shaped section defining said seal joint, and openings of the C-shape and the reverse C-shape defining said slot.

8. The segmented component of claim **7**, wherein each pair of adjacent segments further comprises adjacent radial extension flanges extending substantially perpendicular to said gas path, said seal joint being disposed at a distal end of said adjacent radial extension flanges.

9. The segmented component of claim **8**, wherein said adjacent radial extension flanges have a length substantially corresponding to the distance between said seal joint and said gas path.

10. The segmented component of claim **8**, further comprising impingement plates secured at one end to said seal joint and extending substantially parallel to said adjacent radial extension flanges and then substantially parallel to said gas path.

11. The segmented component of claim **10**, further comprising at least one flow slot defining a coolant flow path between an area adjacent said impingement plates and said gas path.

12. The segmented component of claim **10**, further comprising a closed circuit cooling region defined in part by said impingement plates.

13. A segment for a circumferential component of a rotary machine comprising:

a longitudinal section extending in a first direction;

a radial extension flange continuous with said longitudinal section and extending in a second direction substantially perpendicular to said first direction; and

a seal joint section continuous with said radial extension flange and defining a slot, said slot being shaped to receive a sealing member in cooperation with an adjacent slot of an adjacent segment, wherein said seal joint section is radially spaced from said longitudinal section by an amount substantially corresponding to said radial extension flange defining an undercut area between said longitudinal section and said seal joint section.

14. A method of sealing and cooling circumferentially adjacent segments in a gas turbine engine, the method comprising:

locating seal joint sections, including seal joint slots in the adjacent segments, radially spaced from a gas path by an amount sufficient to enable active cooling of the seal joint slots; and

inserting a sealing member in the seal joint slots of adjacent segments to seal the adjacent segments.

15. A method according to claim **14**, further comprising one of open-circuit impingement cooling and convection cooling of the seal joint slots.

16. A method according to claim **15**, further comprising film cooling of the seal joint slots.

17. A method according to claim **14**, further comprising one of closed-circuit impingement cooling and convection cooling of the seal joint slots.

* * * * *

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(12) **United States Patent**
Anderson et al.

(10) **Patent No.: US 6,261,053 B1**
(45) **Date of Patent: Jul. 17, 2001**

(54) **COOLING ARRANGEMENT FOR GAS-TURBINE COMPONENTS**

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(73) Assignee: **Asea Brown Boveri AG**, Baden (CH)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/151,809**

(22) Filed: **Sep. 14, 1998**

(30) **Foreign Application Priority Data**

Sep. 15, 1997 (EP) 97810659

(51) **Int. Cl.**[7] **F01D 5/14**
(52) **U.S. Cl.** **415/115**
(58) **Field of Search** 415/115, 134, 415/136, 137, 138, 139, 208.2

(56) **References Cited**

U.S. PATENT DOCUMENTS

3,801,220	*	4/1974	Beckershoff	416/198
3,975,114	*	8/1976	Kalkbrenner	415/138 X
4,311,432	*	1/1982	Kildea	415/115
4,650,394	*	3/1987	Weidner	415/115
4,902,198	*	2/1990	North	415/115
5,020,970	*	6/1991	Dussourd et al.	415/115
5,088,888	*	2/1992	Bobo	415/134 X
5,167,485	*	12/1992	Starkweather	415/115
5,374,161	*	12/1994	Kelch et al.	415/139
5,823,741	*	10/1998	Predmore et al.	415/134

FOREIGN PATENT DOCUMENTS

2280935A		2/1995	(GB) .	
60-22002	*	2/1985	(JP)	415/139
WO94/12765		6/1994	(WO) .	

* cited by examiner

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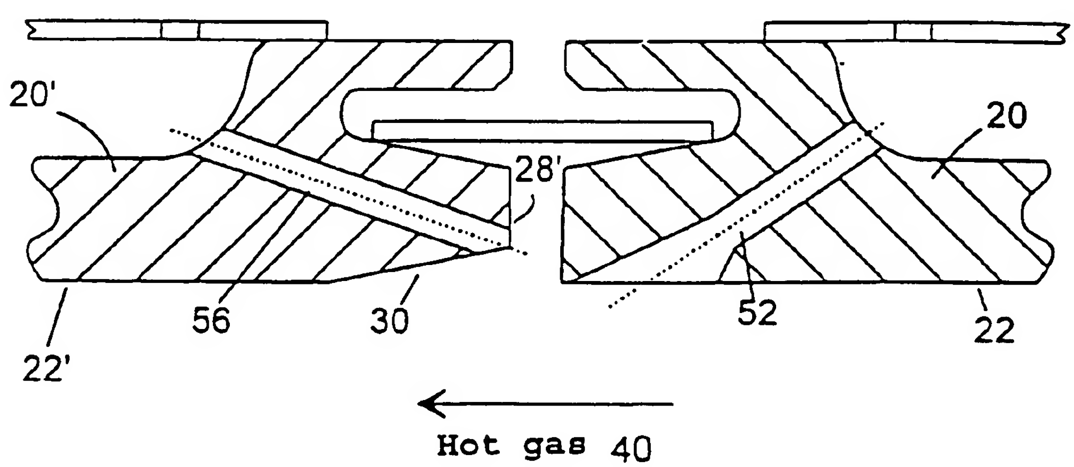
(74) *Attorney, Agent, or Firm*—Burns, Doane, Swecker & Mathis, L.L.P.

(57) **ABSTRACT**

A hot-gas stream (40) flows along the surface of a segment arrangement for shroud bands, in particular in a gas turbine. The segment arrangement comprises segments (20, 20') arranged next to one another and in each case separated from one another by a gap (12). The hot-gas stream (40), in at least one section (70) of the gap (12), has a velocity component perpendicular to the direction of the gap from a first segment (20) to a second segment (20'). In this case, in said section (70), along that edge (26) of the first segment (20) which faces the gap (12), at least one film-cooling bore (52) connects a cooling-air chamber (50), allocated to the first segment, to the surface (22) subjected to the hot-gas stream (40), and/or, in said section (70), along that edge (26') of the second segment (20') which faces the gap (12), at least one edge-cooling bore (56) connects a cooling-air chamber (50'), allocated to the second segment, to the inside (28') of the gap (12).

23 Claims, 8 Drawing Sheets



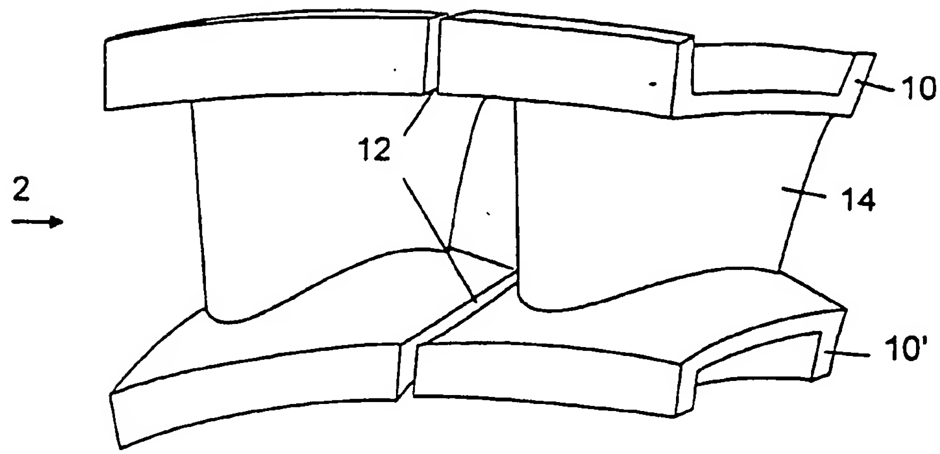


Fig. 1

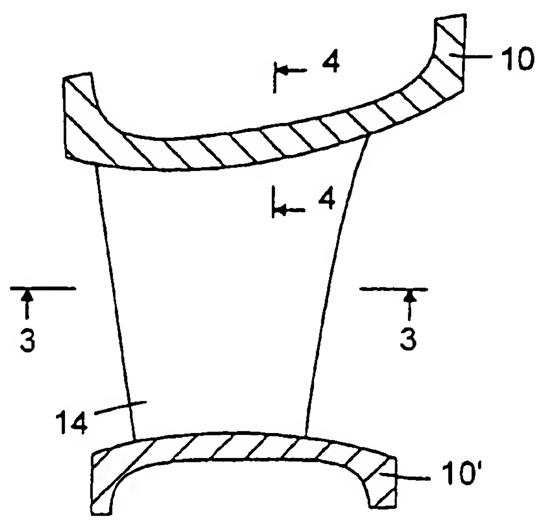


Fig. 2

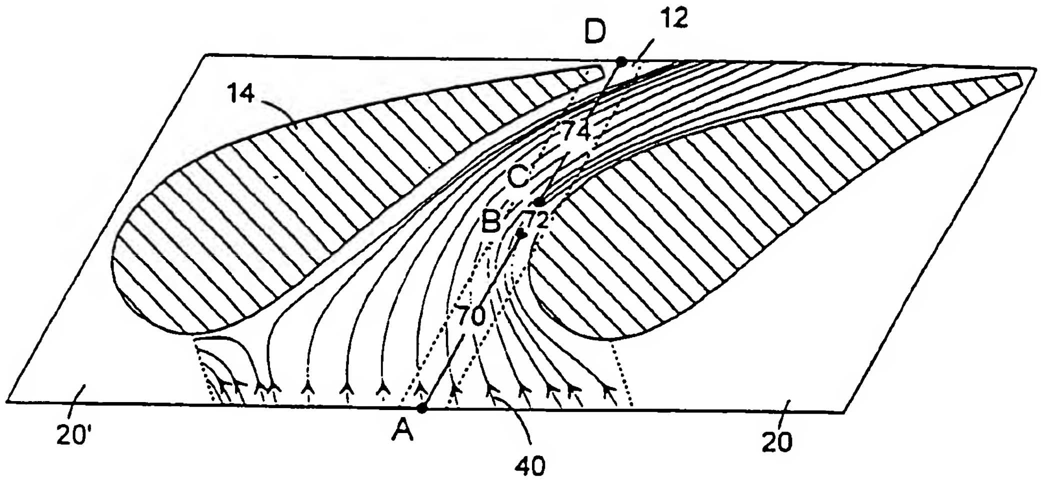


Fig. 3

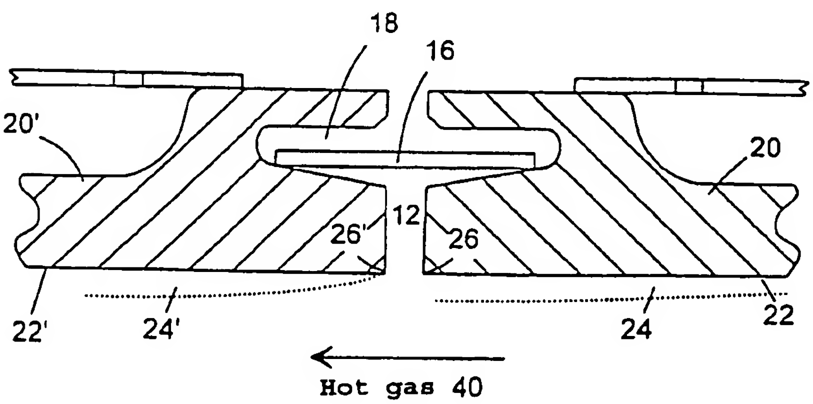


Fig. 4 (Prior art)

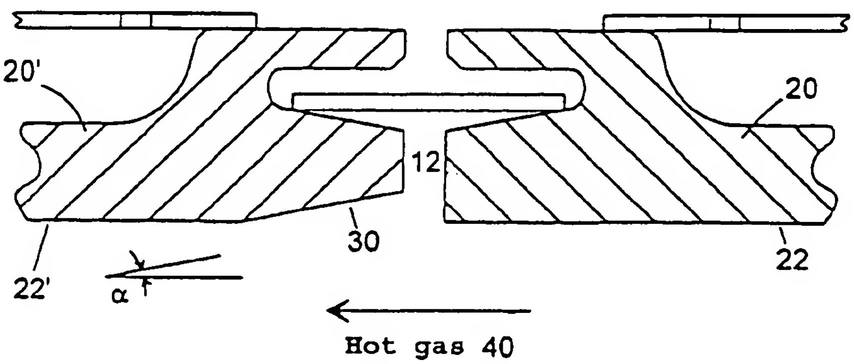


Fig. 5

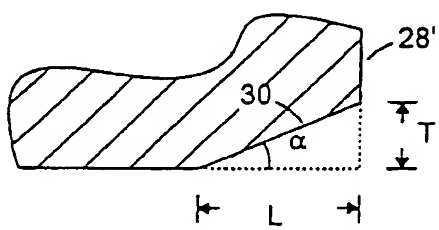


Fig. 6a

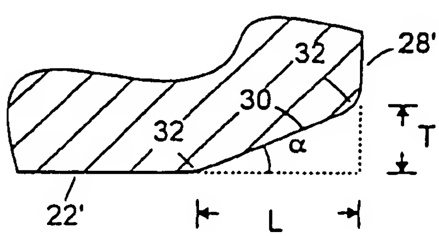


Fig. 6b

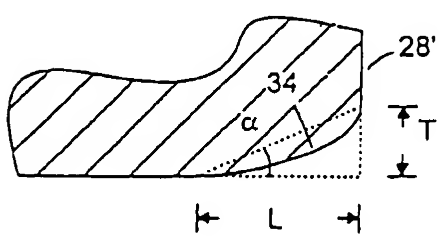


Fig. 6c

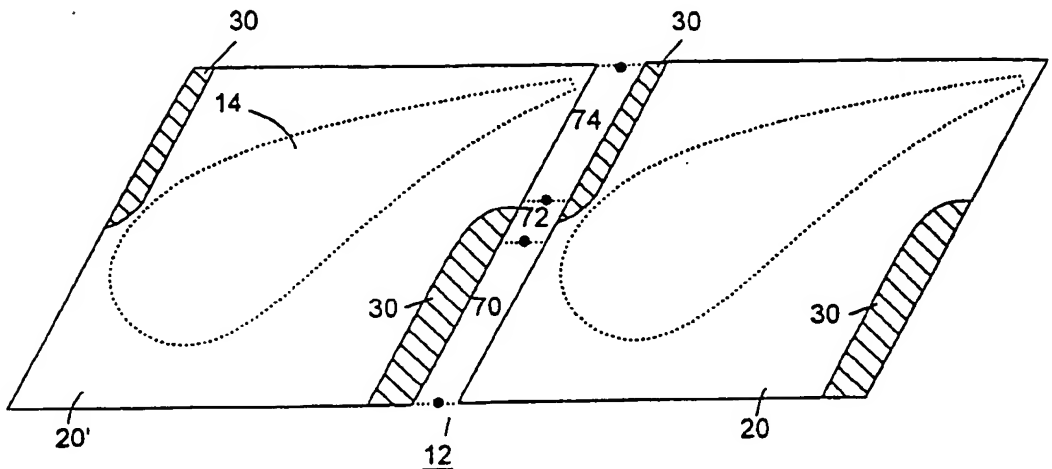


Fig. 7

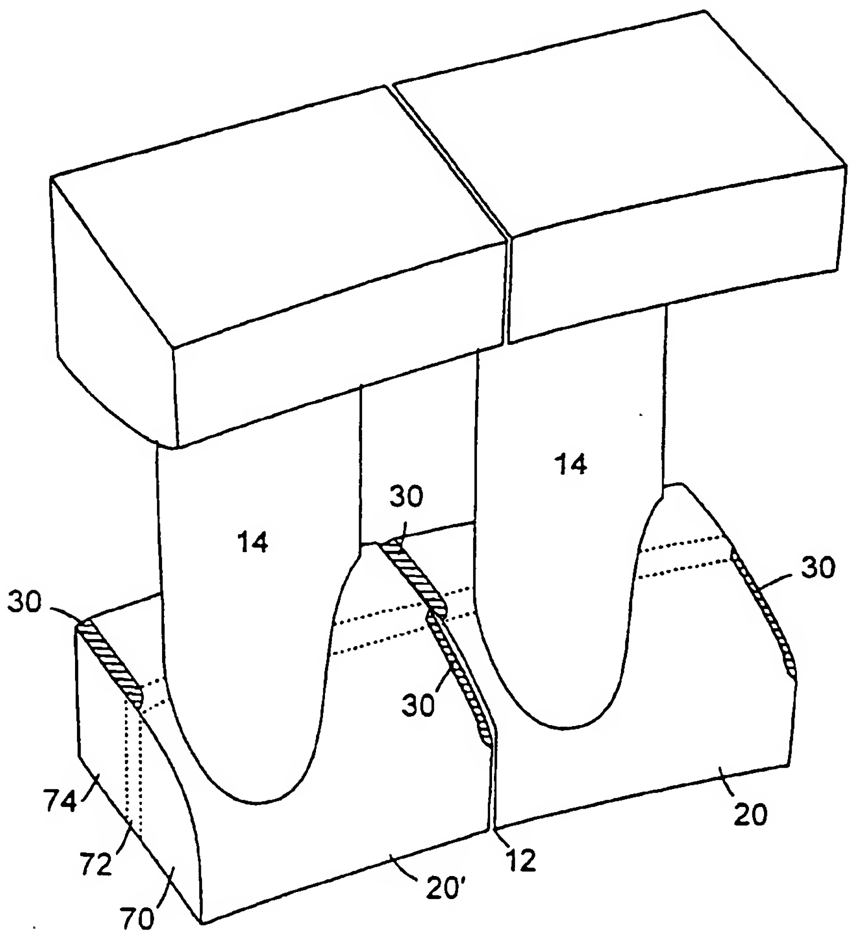


Fig. 8

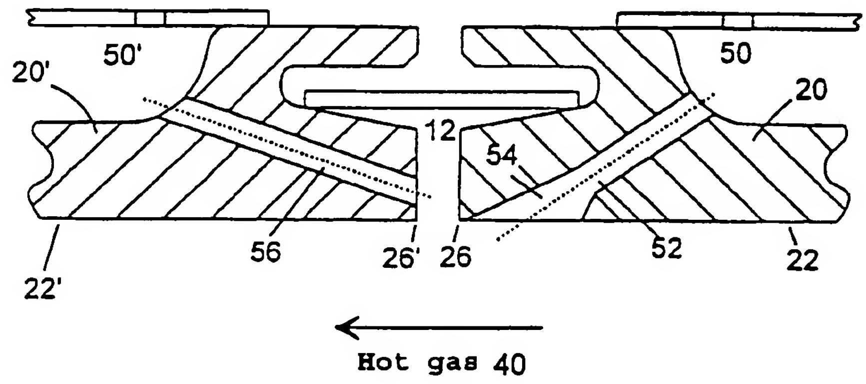

50'
20'
50
20
12
54
56
26'
26
52
22
22'
Hot gas 40

Fig. 9

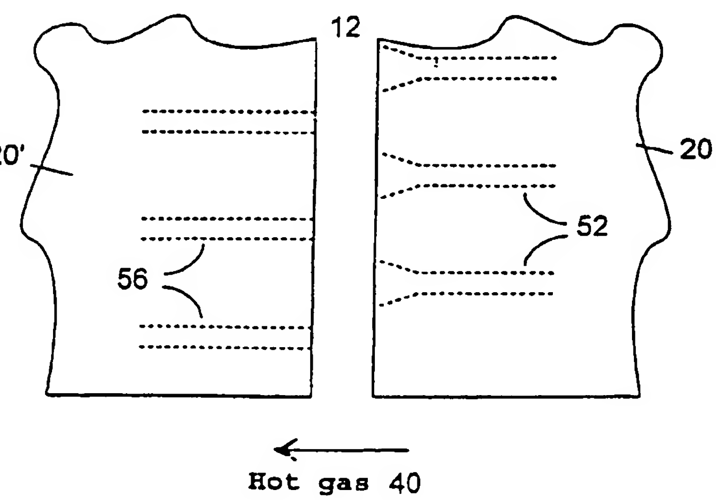

12
20'
20
52
56
Hot gas 40

Fig. 10

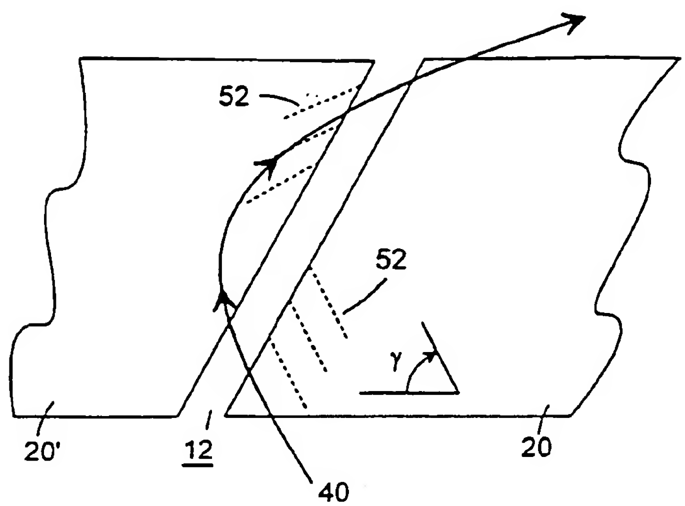


Fig. 11

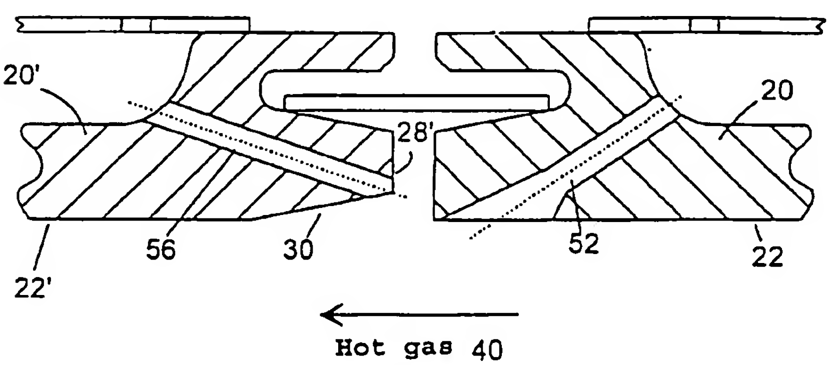


Fig. 12

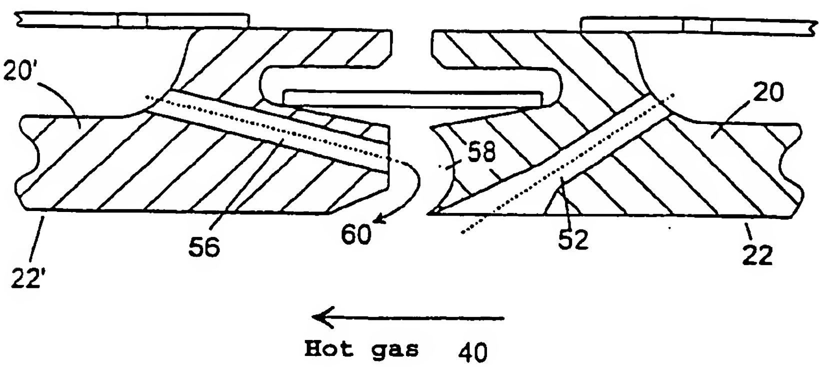


Fig. 13

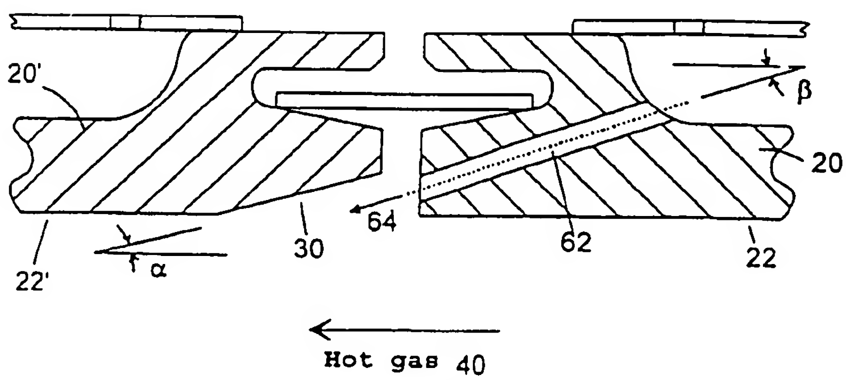


Fig. 14

COOLING ARRANGEMENT FOR GAS-TURBINE COMPONENTS

FIELD OF THE INVENTION

The invention relates to a segment arrangement for platforms, in particular in a gas turbine, along the surface of which a hot-gas stream flows, having segments arranged next to one another and in each case separated from one another by a gap, the hot-gas stream, in at least one section of the gap, having a velocity component perpendicular to the direction of the gap from a first segment to a second segment.

BACKGROUND OF THE INVENTION

In order to achieve a maximum turbine output, it is advantageous to work at the highest possible gas temperatures. In modern gas turbines, the temperatures are so high that many components have to be cooled, since otherwise the temperature of the components which is permissible for maximum durability would be exceeded. A suitable design and/or cooling of critical components is therefore of crucial importance in modern gas turbines. The cooling problem of platforms occurs to an increased extent in annular combustion chambers, since the latter produce a very uniform temperature profile at the entry to the turbine. This means that the platform of the blade has to bear almost the average hot-gas temperature. To achieve the lowest possible NOx emissions, virtually the entire proportion of the combustion air is delivered through the burners themselves in modern combustion chambers; the proportion of the cooling air for the film cooling of the combustion chamber is therefore reduced. This likewise leads to a more uniform temperature profile at the turbine entry and thus to increased thermal loading.

Critical components in turbines are, inter alia, heat shields, combustion-chamber segments and combustion-chamber plates, moving and guide blades, inner and outer shroud bands of the moving and guide blades, and also moving- and guide-blade platforms.

In particular at the sides of segments (platforms) arranged next to one another, experience shows that increased thermal loading often occurs. If, for instance, the segments of a platform are coated with a heat-insulating coating, peeling of the coating is often found. This results in a weak point, at which oxides rapidly form, and these oxides in turn encourage the peeling of the coating. Large uncoated metal surfaces can thus be subjected to the hot-gas stream in a short time.

SUMMARY OF THE INVENTION

Accordingly, one object of the invention is to design the components which are critical with regard to high temperatures, in particular components composed of segments, in such a way that the thermal loading of these components is effectively reduced with the simplest possible means.

This object is achieved according to the invention in a first aspect in that that edge of each segment which is subjected to the hot-gas stream is set back from the impinging hot-gas stream.

The hot-gas stream flows along the surface of the segments, which are arranged next to one another and are in each case separated from one another by a gap. It can now be found that the boundary layer separates at the gap located between the platforms arranged next to one another. The

boundary layer then forms again at that side edge of the downstream platform which is acted upon by the hot-gas stream. Very high heat transfer and thus increased thermal loading of the exposed material therefore occur at this point due to the very thin boundary layer. If there are protruding steps at the gap due to production tolerances, the material of these steps is subjected to especially high thermal loading by the impinging hot-gas stream. Furthermore, hot gas can be deflected into the gap by the edges and in particular by projecting steps. This may lead to a reduced service life and also often to damage to the components adjoining the gap.

A remedy is provided here by that edge of each segment which is acted upon by the hot-gas stream being set back according to the invention by bevelling or rounding off the edge. In this case, the edge need not be bevelled or rounded off over its entire length, since the direction of the hot-gas stream on the segment can change. A precondition for the applicability of the invention, however, is that the hot-gas stream, in at least one section of the gap between the segments, has a velocity component perpendicular to the direction of the gap and thus points from a first to a second segment. In this section, that edge of the second segment which faces the gap is acted upon by the hot-gas stream and is therefore rounded off or bevelled in the first aspect of the invention.

In the case just described, the hot gas flows over the gap, which separates two segments, from the surface of the first segment to the surface of the second segment. In this case, the main component of the velocity of the hot-gas stream is mostly directed from the front side to the rear side of the segments, that is, along the gap. In addition to this flow across the segments, the hot-gas stream has a velocity component perpendicular or at right angles to the gap. Only this perpendicular velocity component leads to the segment edges being acted upon by the hot-gas stream. In many applications, the hot-gas stream on the second segment changes its direction, for instance owing to the fact that a guide device or an airfoil part is put on each segment. In this case, the velocity component along the gap is largely retained; the velocity component perpendicular to the gap merely reverses. The result of this is that there is then a second section downstream of the first section in which the hot-gas stream flows from the first to the second segment, in which second section the hot-gas stream flows from the second segment over the gap to the surface of the first segment. In this second section, that edge of the first segment which faces the gap is then advantageously rounded off or bevelled, since in this section this edge is acted upon by the hot-gas stream.

In a transition region between the first and the second section, the hot-gas stream will flow essentially parallel to the direction of the gap. It is now advantageous if the bevels or rounded-off portions of the first and second sections respectively are gradually reduced to zero in this transition region.

The segment arrangements described here generally involve a multiplicity of segments which are arranged next to one another, so that in each case two segments are separated from one another by a gap. The segment arrangement as a unit may form, for example, a closed ring or it may be arranged on the inside or outside on the circumference of a cylinder. As a rule, the segments are identical, except for any end pieces, so that it always suffices in the present invention to describe a single segment. If a first and a second segment are referred to in the present invention, this relates to two segments which are selected as an example and lie on the two sides of a gap. This serves to illustrate the direction

of the gas flow and does not mean that the invention is restricted to only two segments. The gap between two segments may vary statistically due to production tolerances, and in the extreme case may even be omitted at individual segments as a result.

Within the scope of the first aspect of the present invention, the edges are bevelled at an angle α of between 1 and 60 degrees, preferably between 20 and 40 degrees, and in particular preferably less than about 30 degrees. If the bevel extends over a length L perpendicular to the gap, the depth T of the bevel is related to the length L and the angle α via tan α=T/L. In general, the maximum depth is predetermined, for instance by a recess in the interior of the segments, in which recess a sealing strip, for example, is located. It has been found that, with this selection of the angle, a depth of less than the maximum depth and a corresponding length of the bevel, the risk of separation of the boundary layer at the gap is markedly reduced.

It is also advantageous for the stability of the boundary layer if the transition points between the bevelled region and the inside of the gap, and/or between the bevelled region and the unaltered surface of the segment are not of abrupt design but are rounded off, for instance in the form of an elliptical section.

As an alternative to the bevel, the entire set-back portion of the edge may also be effected by a rounded-off portion, advantageously in the form of an elliptical section. The projection of the rounded-off portion to the front side then shows a quarter ellipse, the ellipse having semiaxes of length L and T respectively. It has also been found here that the risk of separation of the boundary layer at the gap is markedly reduced if L and T are selected in such a way that the angle α=arc tan (T/L) is between 1 and 60 degrees, preferably between 20 and 40 degrees, in particular preferably around about 30 degrees. Here, too, the depth T is limited by the maximum depth.

Production tolerances during the manufacture of the segments are also to be taken into account when selecting the depth T. The predetermined depth T is advantageously selected in such a way that, when the tolerances are taken into account, at least a large percentage, for example more than 50%, of all segments are set back by the bevel or rounded-off portion.

In accordance with the above explanations, in each case a certain value, e.g. T1 in the first section and T2 in the second section, is selected for the depth T in the first and the second sections respectively in case the hot-gas stream on the segment changes its direction. Both values are selected so as to be smaller than the maximum depth. In the transition region, T1 and T2 are now gradually reduced to zero. This may be effected linearly or advantageously in an arc of a circle or an ellipse.

Although the first aspect of the invention already leads to a significant reduction in the thermal loading of the segment edges, for components having very high thermal loading, such as, for instance, the inlet guide blades in gas turbines, further measures for achieving the maximum service life of the components are often advantageous. Thus the object is achieved according to the invention in a second aspect in that a suitable arrangement of film-cooling bores and/or edge-cooling bores in the segment directs cooling air from a cooling-air chamber to the surface subjected to the hot-gas stream.

In this aspect of the invention, a cooling-air chamber is advantageously allocated to each segment. In this case, a different cooling-air chamber may be provided for each segment, but the same cooling-air chamber may also be allocated to a plurality of segments. Again, a precondition for the applicability of the invention is that the hot-gas stream, in at least one section of the gap between the segments, has a velocity component perpendicular to the direction of the gap and thus points from a first to a second segment. In this section, at least one film-cooling bore is made along that edge of the first segment which faces the gap. These film-cooling bores may have any shape; they are preferably cylindrical and/or funnel-shaped and in particular preferably cylindrical on the side of the cooling-air chamber and open in a funnel shape toward the hot-gas side. The axis of the film-cooling bores advantageously points not against the direction of flow of the hot gas but toward the gap and encloses with the surface an angle of 10 to 50 degrees, preferably 25 to 45 degrees, in particular preferably about 35 degrees. As a result, a cooling-air film forms on the surface of the segment in the vicinity of that edge of the adjacent segment which is acted upon by the hot-gas stream, and this cooling-air film cools and thus protects the edge. The action of the cooling film is optimized due to the relatively small angle between bore and surface.

In addition, at least one edge-cooling bore may advantageously be made along that edge of the second segment which faces the gap. The edge-cooling bores begin in the cooling-air chamber allocated to the segment and point toward the gap like the film-cooling bores. The edge-cooling bores may be of any shape; they are preferably of cylindrical shape. Unlike the film-cooling bores of the adjacent segment, the edge-cooling bores do not end at the surface subjected to the hot-gas stream but lead into the gap separating the segments. In this case, care is to be taken to ensure that the angle of the edge-cooling bores is kept so small that the discharging cooling air is not blown immediately into the hot gas, since otherwise excessive losses would occur, since the cooling air is blown out against the direction of flow of the hot gas. The edge-cooling bores enclose with the surface an angle of 5 to 50 degrees, preferably 20 to 40 degrees, in particular preferably about 30 degrees. The purpose of the edge-cooling bores is to provide a cooling-air region in the gap, which cooling-air region acts as a convective heat sink. The edge-cooling bores blow the cooling air into the gap between the segments and thus provide for adequate cooling in the gap. Therefore, the combination of film-cooling bores and edge-cooling bores, on the one hand, delivers a protective cooling-air layer for that side edge of each segment which is acted upon by the hot-gas stream and, on the other hand, provides a heat sink in the separating gap due to the edge-cooling bores, and this heat sink convectively dissipates the heat flow introduced by the hot-gas stream.

Here, too, a second section may be formed in the gap by reversing the direction of flow of the hot-gas stream, in which second section the hot-gas stream flows from the second segment over the gap to the surface of the first segment. In this second section, the arrangement of the film-cooling bores and edge-cooling bores is reversed; at least one film-cooling bore is therefore made there along that edge of the second segment which faces the gap, and at least one edge-cooling bore is made along that edge of the first segment which faces the gap. In a transition region between the first and second sections, the hot-gas stream will flow essentially parallel to the direction of the gap. In this transition region, at least one edge-cooling bore is advantageously made along the edge of each of the two segments which faces the gap. There are therefore no film-cooling bores in the transition region. However, the convective heat extraction provided by the edge-cooling bores is retained for cooling purposes.

If there is a recess, which, for example, contains a sealing strip, in the interior of the segments, the film-cooling bores and edge-cooling bores are made in such a way that they do not intersect this recess.

The film-cooling bores and edge-cooling bores may be made opposite one another at the gap. In a preferred development, however, the film-cooling bores and the edge-cooling bores are staggered laterally, that is, in the direction of the gap. In a further preferred development, the film-cooling bores and/or the edge-cooling bores have a lateral setting angle. The bores are advantageously made with such a setting angle that their axes point approximately in the direction of the hot-gas stream.

In a further embodiment, the first and second aspects of the invention are advantageously combined. The essential features of the comments made about the first and second aspects also apply to the following discussion of this embodiment and the further embodiments. It is also particularly the case that, for the invention, there is at least one section having a hot-gas-stream velocity component perpendicular to the gap, but that there may also be, in addition, a second section having an opposed perpendicular velocity component and a transition region. In this embodiment, film-cooling bores and/or edge-cooling bores are now made as in the second aspect of the invention. In addition, that edge of each segment which is acted upon by the hot gas is bevelled or rounded off, as described in the first aspect of the invention. If edge-cooling bores are provided, the edge-cooling bores and bevel or rounded-off portion, in the one section, and, if present, in the second section, are matched to one another in such a way that the depth of the set-back portion reaches up to the gap-side opening of the edge-cooling bore. The end region of the edge-cooling bores is thereby covered by the bevel or rounded-off portion, as a result of which the edge-cooling bores remain open, even if the gap width should be reduced to zero by production tolerances or transient conditions during operation.

In a further embodiment, film-cooling bores and/or edge-cooling bores are likewise made as in the second aspect of the invention. In addition, that edge of each segment which is acted upon by the hot gas is bevelled or rounded off, as described in the first aspect of the invention. Here, in contrast to the embodiment described previously, the edge-cooling bores and bevel or rounded-off portion are matched to one another in such a way that the depth of the set-back portion does not reach the gap-side opening of the edge-cooling bore. On the other hand, the inside, facing the gap, of the segment opposite the bevelled segment is provided with a concave, for example roughly parabolic, recess, so that the cooling-air stream discharging from the edge-cooling bore is deflected at this recess. In the interior of the edge-cooling bore, the cooling-air stream first of all flows essentially against the direction of the hot-gas stream. Due to the deflection, the cooling-air stream then leaves the gap essentially parallel to the hot-gas stream. This cooling-air stream can thus also form a protective film on the surface.

If there are two sections having an opposed perpendicular velocity component of the hot-gas stream and a transition region in between in the embodiment described, the recess is dispensed with in the transition region. In the transition region, therefore, edge-cooling bores are arranged on both sides of the gap, as described in the second aspect of the invention, and the bevels or rounded-off portions of the respective edges, as described in the first aspect of the invention, are gradually reduced to zero.

In a further embodiment, in the one section of the gap in which the hot-gas stream has a velocity component from a first segment to a second segment, at least one edge-cooling bore is made along that edge of the first segment which faces the gap. These edge-cooling bores form with the surface an angle β and do not end at the surface subjected to the hot-gas stream but lead into the gap separating the segments. As described in the first aspect of the invention, that edge of the second segment which is acted upon by the hot-gas stream is bevelled or rounded off at an angle α. In this case, the angle β is selected to be in a range of around 30 degrees around the angle α. Due to the combination of upstream edge-cooling bore and the downstream set-back portion of the opposite edge, the discharging cooling-air film does not remain restricted to the volume of the gap. On the contrary, it first of all forms a protective cooling-air layer over the bevel or rounded-off portion and then discharges onto the surface around which the hot-gas stream flows. On account of the selected orientation of the edge-cooling bore and bevel, the cooling-air stream discharges essentially parallel to the direction of the hot-gas stream, as a result of which an optimum cooling-air film is produced. This embodiment, too, offers the advantage that the edge-cooling bores remain open, even if the gap width should be reduced to zero by production tolerances and/or transient conditions during operation.

All aspects and embodiments described may be combined with one another within the scope of the invention in order to reduce the thermal loading of critical components as far as possible. Even though the invention has been described above for one segment arrangement, in particular for a gas turbine, the invention may nonetheless be advantageously used in the case of all critical components which are composed of segments and are subjected to a hot-gas stream, such as, for example, heat shields, combustion-chamber segments and combustion-chamber plates, moving and guide blades, inner and outer shroud bands of the moving and guide blades, and also moving- and guide-blade platforms. Use is not restricted to gas turbines; other hot-gas systems, for instance aircraft turbines, are also within the scope of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

A more complete appreciation of the invention and many of the attendant advantages thereof will be readily obtained as the same becomes better understood by reference to the following detailed description when considered in connection with the accompanying drawings, wherein:

FIG. 1 shows a schematic perspective view of the guide-blade platforms of a gas turbine;

FIG. 2 shows a side view of the guide-blade platform from direction 2 in FIG. 1;

FIG. 3 shows a bottom view of the bottom platform from direction 3—3 in FIG. 2 with depiction of the particle trajectories of the hot-gas stream;

FIG. 4 shows a section of the connecting point between two platform segments in direction 4—4 in FIG. 2 (prior art);

FIG. 5 shows a section like FIG. 4 in a first exemplary embodiment of the invention;

FIG. 6*a* shows a detail view of FIG. 5 in the region of the bevel/rounded-off portion of the downstream segment;

FIG. 6*b* shows a detail view of a modification of FIG. 5 in the region of the bevel/rounded-off portion of the downstream segment;

FIG. 6*c* shows a detail view of a further modification of FIG. 5 in the region of the bevel/rounded-off portion of the downstream segment;

FIG. 7 shows a bottom view like FIG. 3 in the first exemplary embodiment of the invention;

FIG. 8 shows a perspective view of two segments of a guide blade of a high-pressure turbine in the first exemplary embodiment of the invention;

FIG. 9 shows a section like FIG. 4 in a second exemplary embodiment of the invention;

FIG. 10 shows a schematic top or bottom view of a section of two segments in accordance with a development of the second exemplary embodiment of the invention;

FIG. 11 shows a schematic top or bottom view of two segments in accordance with a further development of the second exemplary embodiment of the invention;

FIG. 12 shows a section like FIG. 4 in a third exemplary embodiment of the invention;

FIG. 13 shows a section like FIG. 4 in a fourth exemplary embodiment of the invention;

FIG. 14 shows a section like FIG. 4 in a fifth exemplary embodiment of the invention.

Only the elements essential for the understanding of the invention are shown. Not shown are, for example, the complete guide blade ring, the combustion chamber and the exhaust-gas casing of the gas turbine with exhaust-gas tube and stack.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings, wherein like reference numerals designate identical or corresponding parts throughout the several views, FIGS. 1 to 4 show various views of a guide blade of a gas turbine having an annular combustion chamber according to the prior art. The guide blade comprises a multiplicity of top and bottom platforms 10 and 10' and in each case an airfoil piece 14 arranged in between. In each case two of the top and bottom platforms (generally: segments) are separated from one another by a gap 12.

FIG. 3 shows a bottom view of two top platforms 10 lying next to one another. All explanations equally apply to top and bottom platforms; the platforms are therefore generally designated as segments 20, 20'. In FIG. 3, the particle flow lines of the hot-gas stream 40 are depicted on the surface of the segments. These flow lines are obtained by computer simulation or by direct measurement of worn-out components. The hot-gas stream 40 essentially has a velocity component along the gap 12. In addition, there is a transverse component (also: perpendicular velocity component), which results in the hot-gas stream leading from one segment across the gap to the adjacent segment. In FIG. 3, the transverse component reverses its sign due to the effect of the airfoil piece 14. In one section 70 of the gap, between the points A and B, the hot-gas stream flows from the first segment 20 toward the second segment 20'. In a second section 74 of the gap, between the points C and D, the reverse is the case, i.e. the hot-gas stream flows from the second segment 20' to the first segment 20. In a transition region 72, the direction of the hot-gas stream is essentially parallel to the direction of the gap 12.

FIG. 4 shows a section of the connecting point between two segments in detail. The recess 18 arranged in the interior of the segments 20 and 20' contains a sealing strip 16. The hot-gas stream 40 flows along the surfaces 22 and 22' of the segments 20 and 20'. In this case, the main component of the hot-gas stream in FIG. 4 goes into the plane of the paper. In addition, there is in general a transverse direction, which in FIG. 4 is identified by an arrow. With the specified orientation, FIG. 4 shows a section in the first section 70 (between the points A and B in FIG. 3). If the transverse component points approximately as in FIG. 4 from the first segment 20 to the second segment 20', experience shows that the boundary layer 24 separates at the gap 12. Although it forms again at the second segment 20' (reference numeral 24'), it is very thin directly at the edge 26'. Consequently, increased thermal loading of the segment material occurs in the region of the edge 26'.

FIG. 5 shows a first exemplary embodiment in accordance with the first aspect of the invention. In the first exemplary embodiment, in the case of a transverse component from the first segment 20 to the second segment 20', the edge acted upon by the hot-gas stream is set back by bevelling at an angle of about 30 degrees (reference numeral 30). It has been found that, as a result, the boundary layer 24 of the first segment 20 situated upstream does not separate at the gap 12. The depth T of the bevel is selected in such a way that the bevel does not extend up to the recess 18 containing the sealing strip 16. The bevel 30 as just described is shown in FIG. 6*a*; alternative modifications are shown in detail in FIG. 6*b* and 6*c*.

In FIG. 6*b*, the transition between the bevelled surface 30 and the inside of the gap 28' is not made abrupt but is rounded off, for instance in the form of an elliptical section 32. The same applies to the transition between the bevelled surface 30 and the unaltered surface of the segment 22'. The risk of separation of the boundary layer is further reduced by this development. FIG. 6*c* shows that the entire set-back portion can also be made as a rounded-off portion 34. This is advantageously done in the form of a quarter ellipse as in FIG. 6*c*. The ellipse has semiaxes of length L and T respectively. An angle comparable with the bevel of FIG. 6*a* is obtained in the case of the elliptical section by the ratio of the two semiaxes. In FIG. 6*c*, this angle, $\alpha = \arctan (T/L)$, is selected to be the same size as in FIG. 6*a*, and is therefore 30 degrees here. The risk of separation of the boundary layer at the gap is also very small in this alternative.

According to FIG. 3, the transverse component of the velocity of the hot-gas stream in the present exemplary embodiment reverses its sign due to the effect of the airfoil piece 14. In the first section 70 of the gap, in which the hot-gas stream flows from the first segment 20 toward the second segment 20', that edge of the second segment 20' which is acted upon by the hot-gas stream is bevelled. FIGS. 7 and 8 show that, in the second section 74 of the gap, in which the hot-gas stream flows from the second segment 20' to the first segment 20, that edge of the first segment 20 which is acted upon by the hot-gas stream is bevelled. In a transition region 72 in which the direction of the hot-gas stream is essentially parallel to the direction of the gap 12, the depth of the bevel on both segments is gradually reduced to zero.

A second exemplary embodiment in accordance with the second aspect of the invention is shown in FIG. 9. The section of FIG. 9 shows the case where the transverse component of the velocity of the hot-gas stream points from the first segment 20 to the second segment 20'. In the section of the gap having this orientation, a plurality of film-cooling bores 52 are made along that edge 26 of the first segment 20 which faces the gap 12, and these film-cooling bores 52 connect the allocated cooling-air chamber 50, located here on the rear side of the segment 20, to the surface 22 subjected to the hot-gas stream. The cylindrical film-cooling bores 52 open in a funnel shape (reference numeral 54) toward the hot-gas side 22. The axis of the film-cooling

bores 52 points toward the gap 12 and encloses an angle here of about 35 degrees with the surface 22.

A plurality of edge-cooling bores 56 are made along that edge 26' of the second segment 20' which faces the gap 12. They are of cylindrical shape and point toward the gap 12 at an angle of about 30 degrees. The edge-cooling bores 56 do not end at the surface 22' subjected to the hot-gas stream but lead into the gap 12. The edge-cooling bores 56 provide cooling air in the gap 12, and this cooling air cools down the penetrating hot gas, whereas the film-cooling bores 52 produce a cooling-air film in the vicinity of the edge 26' acted upon by the hot-gas stream, and this cooling-air film cools and protects the edge 26'. Due to the selected angle between the bores 52, 56 and the surfaces 22, 22', vortices and thus aerodynamic losses are avoided as far as possible. In particular, care is to be taken to ensure that the angle of the edge-cooling bores 56 is kept so small that the discharging cooling air is not blown immediately into the hot gas, since otherwise excessive losses would occur, since the cooling air is blown out against the direction of flow of the hot gas.

As in the first exemplary embodiment, the transverse component of the velocity of the hot-gas stream in the present second exemplary embodiment reverses its sign due to the effect of the airfoil piece 14. In the first section 70 of the gap, in which the hot-gas stream flows from the first segment 20 toward the second segment 20', the arrangement of the film-cooling bores 52 and edge-cooling bores 56 is as described above. In the second section 74 of the gap, in which the hot-gas stream flows from the second segment 20' toward the first segment 20, the arrangement of the film-cooling bores 52 and edge-cooling bores 56 is reversed; the film-cooling bores 52 are therefore made along that edge 26' of the second segment 20' which faces the gap 12, and the edge-cooling bores 56 are made along that edge 26 of the first segment 20 which faces the gap 12. In a transition region 72 between the first and second sections, the hot-gas stream will flow essentially parallel to the direction of the gap. In this transition region 72, in the second aspect of the invention, edge-cooling bores 56 are made along the edge 26, 26' of each of the two segments 20, 20' which faces the gap 12; there are no film-cooling bores 52 there.

The film-cooling bores 52 and the edge-cooling bores 56 are preferably staggered laterally, as shown schematically in the bottom view of FIG. 10. In a further development, the film-cooling bores 52 and the edge-cooling bores 56 have a lateral setting angle γ in such a way that the axes of the bores, in all sections, point approximately in the direction of the hot-gas stream 40. For the sake of clarity, only the film-cooling bores 52 are depicted in FIG. 11, but correspondingly set edge-cooling bores 56 are likewise included in a preferred development.

In a third exemplary embodiment (FIG. 12), the first and second aspects of the invention are advantageously combined. In this case, film-cooling bores 52 and edge-cooling bores 56 are made as in the second exemplary embodiment. In addition, that edge of each segment which is acted upon by the hot gas is bevelled as in the first exemplary embodiment. In this case, the bevel 30 and the edge-cooling bores 56 are matched to one another in such a way that the depth of the bevel 30 reaches up to the opening of the edge-cooling bore 56 on the inside 28' of the gap 12. The gap-side end region of the edge-cooling bores 56 is thereby covered by the bevel 30, and the edge-cooling bores 56 remain open, even if the gap width should be reduced to zero by production tolerances or transient conditions during operation. The configuration of the segments in the individual sections is as described in the first and second exemplary embodiments respectively.

In a fourth exemplary embodiment (FIG. 13), film-cooling bores 52 and edge-cooling bores 56 are likewise made as in the second exemplary embodiment, and that edge of each segment which is acted upon by the hot gas is bevelled as in the first exemplary embodiment. In contrast to the third exemplary embodiment, the depth of the bevel 30 here does not reach the edge-cooling bores 56. In addition, the inside 28 facing the gap 12 is provided with a roughly parabolic recess 58, so that the cooling-air stream 60 discharging from the edge-cooling bore is deflected at the recess 58. Whereas the cooling-air stream 60 in the interior of the edge-cooling bore 56 flows essentially against the direction of the hot-gas stream 40, the cooling-air stream 60, due to the deflection, leaves the gap 12 at the surface 22' essentially parallel to the hot-gas stream 40. As a result, undesirable vortices are avoided as far as possible, and a protective cooling-air film 60 is also delivered by the edge-cooling bores 56. In the second section, film-cooling bores 52 and edge-cooling bores 56 as well as bevel 30 and recess 58 are reversed in a manner analogous to the previous exemplary embodiments. In the transition region, edge-cooling bores 56 are made on both sides of the gap as in the second exemplary embodiment, the depth of the bevels 30 is gradually reduced to zero, and no recesses 58 are made on the insides 28, 28' of the gap.

In a fifth exemplary embodiment (FIG. 14), in the section 70 of the gap in which the hot-gas stream has a velocity component from the first segment 20 to the second segment 20', a plurality of edge-cooling bores 62 are made along that edge 26 of the first segment which faces the gap 12. These edge-cooling bores 62 form with the surface 22 an angle β, here about 40 degrees. The edge-cooling bores 62 do not end at the surface 22 subjected to the hot-gas stream 40 but lead into the gap 12. As in the first exemplary embodiment, that edge 26' of the second segment 20' which is acted upon by the hot-gas stream 40 is bevelled at an angle α, here about 30 degrees. Due to the combination of the edge-cooling bore 62 lying upstream with the bevel 30 lying downstream, the cooling-air stream 64 discharging from the edge-cooling bore 62 does not remain restricted to the volume of the gap 12, but rather forms a protective cooling-air layer over the bevel 30 and then discharges onto the surface 22' around which the hot-gas stream 40 flows. On account of the selected orientation of the edge-cooling bore 62 and bevel 30, the cooling-air stream 64 discharges essentially parallel to the direction of the hot-gas stream, a factor which avoids undesirable vortices and aerodynamic losses and leads to an optimized action of the cooling air.

Obviously, numerous modifications and variations of the present invention are possible in light of the above teachings. It is therefore to be understood that, within the scope of the appended claims, the invention may be practiced otherwise than as specifically described herein.

What is claimed as new and desired to be secured by Letters Patent of the United States is:

1. A segment arrangement for platforms, in particular in a gas turbine, along the surface of which a hot-gas stream flows, the segment arrangement comprising: segments arranged next to one another and in each case separated from one another by a gap, the hot-gas stream, in at least one section of the gap having a velocity component perpendicular to the direction of the gap from a first segment to a second segment, wherein, in said section, that edge of the surface of the second segment which faces the gap is bevelled or rounded off, and that edge of the surface of the first segment which faces the gap is positioned to direct the hot-gas stream above the edge of the second segment.

2. The segment arrangement as claimed in claim 1, wherein, in a second section of the gap, the hot-gas stream has a velocity component perpendicular to the direction of the gap from the second segment to the first segment, and, in said section, that edge of the surface of the first segment which faces the gap is bevelled or rounded off, and that edge of the surface of the second segment which faces the gap is positioned to direct the hot-gas stream above the edge of the first segment.

3. The segment arrangement as claimed in claim 2, including a transition region (72) in which the hot-gas stream flows essentially in the direction of the gap, the transition region being arranged between the first and second sections, and wherein, in said transition region, the bevel or rounded-off portion of the first and second sections is gradually reduced to zero.

4. The segment arrangement as claimed in claim 1, wherein the said edge of the segment being bevelled at an angle α, and α being between 1° and 60°.

5. The segment arrangement as claimed in claim 4, wherein the transition between the bevelled surfaces and the inside of each segment is rounded off in the form of an elliptical section.

6. The segment arrangement as claimed in claim 1, wherein the said edges of the segments are rounded off in the form of a quarter ellipse, and the ellipse having semiaxes of length L and T respectively, the angle α=arc tan T/L being between 1° and 60°.

7. The segment arrangement as claimed in claim 1, wherein a sealing strip is arranged in a recess in the gap, and the depth of the bevel or rounded-off portion always being selected to be so small that the bevel or rounded-off portion does not reach the recess.

8. The segment arrangement as claimed in claim 1, in which, each case in that section of a segment which, at the gap, is opposite that section of a further segment which is provided with a bevel, at least one edge-cooling bore, along that edge of the said segment which faces the gap, connects a cooling-air chamber, allocated to the said segment, to the inside of the gap, and in which the axes of the edge-cooling bores enclose with the surface of the said segment an angel β, β lying at an interval of 30° around the angle α defined by the bevel.

9. The segment arrangement as claimed in claim 1, wherein the said edge of the segment is bevelled at an angle α, and α being between 20° and 40°.

10. The segment arrangement as claimed in claim 1, wherein the said edges of the segments are rounded off in the form of a quarter ellipse, and the ellipse having semiaxes of length L and T respectively, the angle α=arc tan T/L is between 20° and 40°.

11. A segment arrangement for platforms for supporting airfoils in a gas turbine in which a hot gas stream flows along the surface of platforms, the segment arrangement comprising: first and second segments arranged adjacent to one another and in each case separated from one another by a gap, the first and second segments having cooling-air chambers, the gap having a first section and a second section, the hot-gas stream at the first section of the gap having a velocity component substantially perpendicular to the direction of the gap when the hot gas stream is flowing from the first segment to the second segment, the hot-gas stream having a velocity component perpendicular to the direction of the gap from the second segment to the first segment, and in the second section, along that edge of the second segment which faces the gap, at least one film-cooling bore in the second segment, for conducting cooling air to the surface subjected to the hot-gas stream, and at least one of said segments having at least one edge cooling bore therein along the edge that faces the gap for conducting cooling air from one of the cooling-air chambers to the inside of the gap.

12. A segment arrangement for platforms for supporting airfoils in gas turbine in which a hot gas stream flows along the surface of the platforms, the segment arrangement comprising: first and second segments arranged adjacent to one another and in each case separated from one another by a gap, the first and second segments having cooling-air chambers, the gap having a first section and a second section, the hot-gas stream at the first section of the gap having a velocity component substantially perpendicular to the direction of the gap when the hot-gas stream is flowing from the first segment toward the second segment, and the hot-gas stream at the second section of the gap having a velocity component substantially perpendicular to the direction of the gap from the second segment to the first segment, and, in the first section of the gap along the edge of the first segment which faces the gap, at least one film-cooling bore in the first segment for conducting cooling air from one of the cooling-air chambers toward the surface subjected to the hot gas stream for displacing the gas stream away from the edge of the gap, and, the second section of the gap, along that edge of the second segment which faces the gap, having at least one film-cooling bore for conducting cooling air from one of the cooling-air chambers to the surface subjected to the hot-gas stream for displacing the gas stream away from the edge of the gap, and wherein a transition region in which the hot-gas stream flows essentially in the direction of the gap is arranged between the first and second sections, and wherein in said transition region, along those edges of the two segments which face the gap, at least one edge-cooling bore connects at least one of the cooling-air chambers to the inside of the gap, and at least one of said segments having at least one edge cooling bore therein along the edge that faces the gap for conducting cooling air from one of the cooling-air chambers to the inside of the gap.

13. The segment arrangement as claimed in claim 12, including a plurality of film-cooling bores, and in which the axes of the film-cooling bores point toward the gap and enclose with the surface of the segment containing said film-cooling bores an angle of between 10° and 50°.

14. The segment arrangement as claimed in claim 12, in which in each case that section of the edge which is acted upon by the hot-gas stream is bevelled or rounded off, and in which, in said transition region, the bevel or rounded-off portion of the said sections is gradually reduced to zero.

15. The segment arrangement as claimed in claim 12, in which in each case that section of the edge of each segment which is acted upon by the hot-gas stream in rounded off, the said edges of the segments being rounded off in the form of a quarter ellipse, and the ellipse having semiaxes of length L and T respectively, the said edges of the segments are rounded off in the form of a quarter ellipse, and the ellipse having semiaxes of length L and T respectively, the agleα= are tan T/L are between 1° and 60°.

16. The segment arrangement as claimed in claim 12, wherein in each case that section of the edge of each segment which is acted upon by the hot-gas stream is rounded off, the depth of the rounded-off portions being selected in such a way that they do not reach the openings of the edge-cooling bores at the inside of the gap, and in which the inside of each segment provided with film-cooling bores is provided with a concave recess.

17. The segment arrangement as claimed in claim 12, wherein the axes of the edge-cooling bores point toward the

gap and enclose the surface of the segment containing said edge-cooling bores at an angle of between 5° and 50°.

18. The segment arrangement as claimed in claim **12**, wherein the edge-cooling bores are funnel shaped adjacent the surface of the segment.

19. The segment arrangement as claimed in claim **12**, including a plurality of film-cooling bores, in which the axes of the film-cooling bores point toward the gap and enclose with the surface of the segment containing said film-cooling bores and angle of between 25° and 45°.

20. A segment arrangement for platforms for supporting airfoils in a gas turbine, in which a hot-gas stream flows along the surface of the platforms, the segment arrangement comprising: a plurality of segment arranged adjacent to one another and in each case separated from one another by a gap, the hot-gas stream, in at least one section of the gap, having a velocity component substantially perpendicular to the direction of the gap when the hot-gas stream is flowing from a first segment to a second segment, wherein, said section includes a plurality of film-cooling bores along the edge of the first segment which faces the gap for conducting cooling air to the surface subjected to the hot-gas stream for displacing the gas stream away from the edge of the gap, and in which the film-cooling bores are of cylindrical shape on the side facing the cooling-air chambers and have a funnel-shaped opening on the hot-gas side.

21. A segment arrangement for platforms for supporting airfoils in a gas turbine, in which a hot-gas stream flows along the surface of the platforms, the segment arrangement comprising: a plurality of segments arranged adjacent to one another and in each case separated from one another by a gap, the hot-gas stream, in at least one section of the gap, having a velocity component substantially perpendicular to the direction of the gap when the hot-gas stream is flowing form a first segment to a second segment, wherein, said section includes a plurality of film-cooling bores along the edge of the first segment which faces the gap for conducting cooling air to the surface subjected to the hot-gas stream for displacing the gas stream away from the edge of the gap, and in which the film-cooling bores and edge-cooling bores are from the edge of the gap, and in which the film-cooling bores and edge-cooling bores are staggered laterally.

22. The segment arrangement as claimed in claim **21**, in which the film-cooling bores and the edge-cooling bores have a lateral setting angle γ, and in which the lateral setting angle γ is selected in such a way that the axes of the film-cooling bores of the edge-cooling bores point essentially in the direction of the hot-gas stream.

23. The segment arrangement as claimed in claim **5**, wherein the transition between the bevelled surfaces and the non-bevelled surfaces of each segment is rounded off in the form of an elliptical section.

* * * * *

United States Patent [19]

Hsia et al.

[11] **Patent Number: 4,573,865**

[45] **Date of Patent: Mar. 4, 1986**

[54] **MULTIPLE-IMPINGEMENT COOLED STRUCTURE**

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[73] Assignee: **General Electric Company**, Cincinnati, Ohio

[21] Appl. No.: **595,754**

[22] Filed: **Apr. 2, 1984**

Related U.S. Application Data

[62] Division of Ser. No. 297,688, Aug. 31, 1981, Pat. No. 4,526,226.

[51] **Int. Cl.**[4] **F02C 7/18**; F04D 29/58; F01D 25/12

[52] **U.S. Cl.** .... **415/115**; 165/109.1; 165/908; 415/219 R; 416/97 R

[58] **Field of Search** .... 415/115, 116, DIG. 1, 415/219 R; 416/97 R, 97 A; 165/109, DIG. 11, 109 R

[56] **References Cited**

U.S. PATENT DOCUMENTS

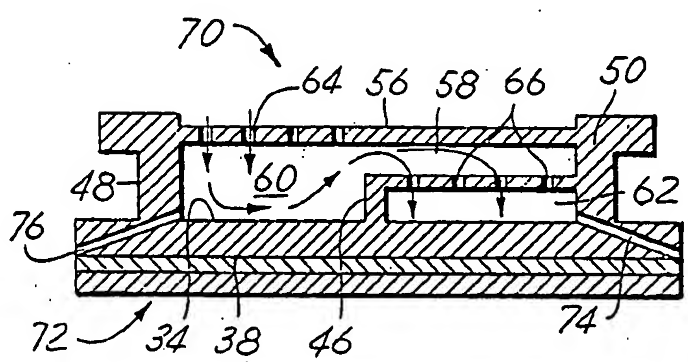
3,348,622	10/1967	Papst	415/DIG. 1
3,597,102	8/1971	Unsworth et al.	415/DIG. 1
3,728,039	4/1973	Plemmons et al.	415/115
3,825,364	7/1974	Halila et al.	415/116
3,975,901	8/1976	Hallinger et al.	415/115 X
4,177,004	12/1979	Riedmiller et al.	415/116
4,465,429	8/1984	Martin et al.	415/115 X

Primary Examiner—Sheldon J. Richter
Attorney, Agent, or Firm—Gregory A. Welte; Derek P. Lawrence

[57] **ABSTRACT**

A multiple-impingement cooled structure, such as for use as a turbine shroud assembly. The structure includes a plurality of baffles which define with an element to be cooled, such as a shroud, a plurality of cavities. Impingement cooling air is directed through holes in one of the baffles to impinge upon only the portion of the shroud in a first cavity. That cooling air is then directed to impinge again upon the portion of the shroud in a second cavity.

4 Claims, 5 Drawing Figures



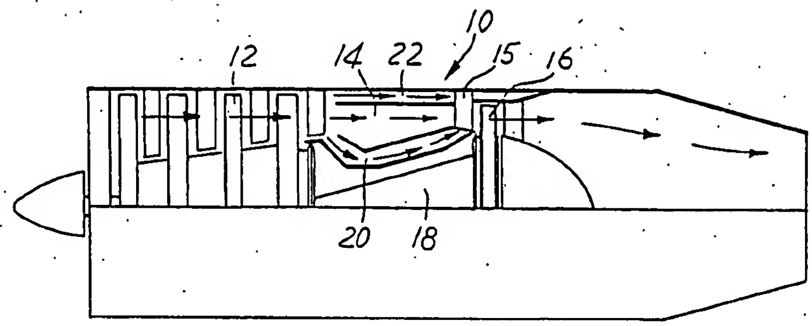


Fig 1

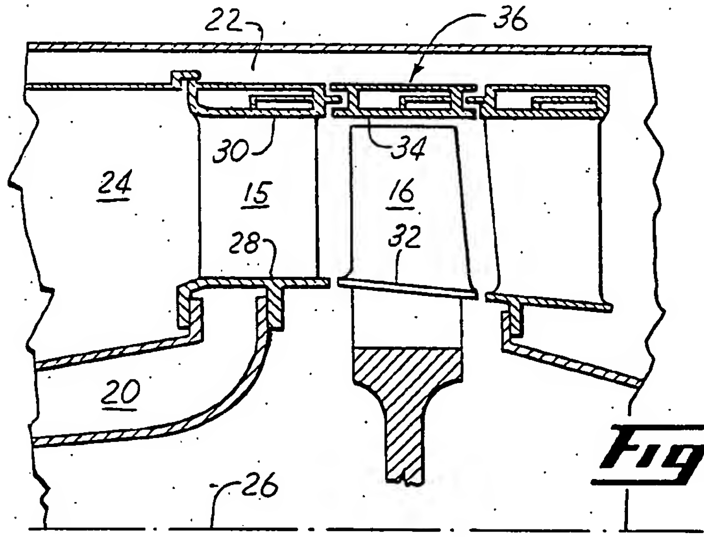


Fig 2

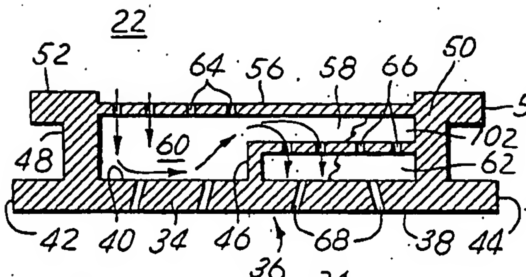


Fig 3

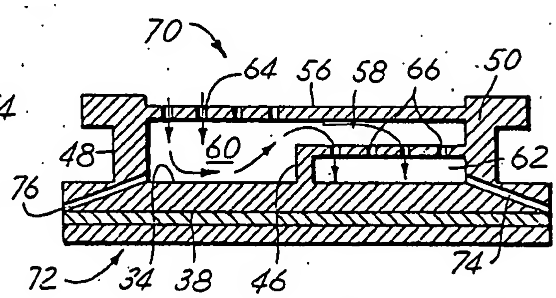


Fig 4

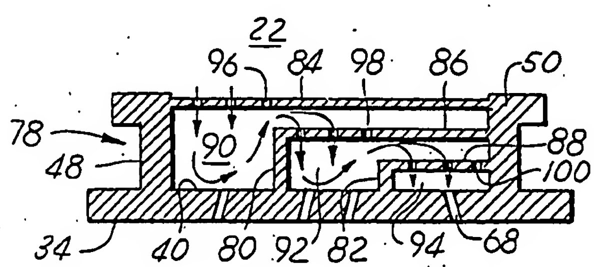


Fig 5

MULTIPLE-IMPINGEMENT COOLED STRUCTURE

This is a division of application Ser. No. 297,688, filed Aug. 31, 1981, now U.S. Pat. No. 4,526,226.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to structural cooling and particularly to a new and improved multiple-impingement cooled structure, such as for use as a turbine shroud assembly.

2. Description of the Prior Art

Structures, such as turbine shrouds and nozzle bands, which are subjected to high temperatures must be cooled in order to reduce possible damage caused by undesirable thermal expansion and to maintain satisfactory sealing characteristics. Several methods of cooling such structures are currently being successfully employed.

One method is film cooling. In film cooling, a thin film of cooling fluid, such as air, is directed to flow along and parallel to the surface which is to be cooled. Although film cooling provides excellent cooling, when used adjacent a gas stream, such as along the inner surface of a turbine shroud in the turbine section of an engine, the film cooling air mixes with the gases in the gas stream. The momentum of the film cooling air is lower than the momentum of gases with which it mixes and thus the resultant overall momentum of the mixed gas stream is lowered. Also, the mixing of the film cooling air with the gases in the gas stream imparts some turbulence to the gas stream. The net result of the mixing of the film cooling air with the gas stream is, in the case of the turbine section of an engine, that there is less work available to rotate the turbine rotor and thus turbine efficiency is decreased. Correspondingly, the greater the amount of film cooling air used, the greater will be the turbine efficiency decrease caused by mixing losses.

Another method of cooling structures is impingement cooling. In impingement cooling, air is directed to impinge substantially perpendicularly upon the surface of a structure to be cooled. When used on a turbine shroud, for example, cooling air is directed to impinge upon the back or outer surface of the shroud, that is, the surface not facing the gas flowpath. The source of the cooling air for both impingement and film cooling air in most gas turbine engines is high pressure air from the compressor. For effective impingement cooling of the entire turbine shroud in current impingement cooling arrangements, a relatively large amount of cooling air must be employed and thus the compressor must work harder to supply the cooling air. Thus, when a large amount of cooling air is required for impingement cooling, engine efficiency is reduced.

In view of the above-mentioned problems, it is therefore an object of the present invention to provide a structure having a unique configuration whereby it can be satisfactorily cooled with a reduced amount of film cooling air to thereby reduce mixing losses.

Another object of the present invention is to provide a structure configured whereby impingement cooling air is directed to impinge more than once upon an element of the structure to be cooled, thus requiring a reduced amount of cooling air and thereby increasing engine efficiency.

BRIEF DESCRIPTION OF THE DRAWING

This invention will be better understood from the following description taken in conjunction with the accompanying drawing, wherein:

FIG. 1 is a view of the upper half of a gas turbine engine with a portion cut away to show some engine components therein.

FIG. 2 is a cross-sectional view of a portion of the turbine section of a gas turbine engine incorporating features of the present invention.

FIG. 3 is a cross-sectional view of one embodiment of a shroud assembly of the present invention.

FIG. 4 is a cross-sectional view of another embodiment of the shroud assembly of the present invention.

FIG. 5 is a cross-sectional view of yet another embodiment of the shroud assembly of the present invention.

SUMMARY OF THE INVENTION

The present invention comprises a multiple-impingement cooled structure. The structure comprises an element to be cooled and a plurality of baffles having impingement holes therethrough. The baffles partially define with portions of the element a plurality of cavities. The baffles and cavities are arranged for directing cooling fluid from a source thereof to impinge sequentially upon the portion of the element within each of the cavities. The structure also includes fluid communication means between at least one of the cavities and the exterior of the structure.

In a particular embodiment of the structure of the present invention, the element which is to be cooled includes flanges near the ends thereof and a rib between the flanges. A first baffle extends between the flanges and a second baffle extends between the rib and a flange. Cooling air is directed to impinge upon the portion of the element in a first cavity and then upon the portion of the element in a second cavity.

In another embodiment of the invention, the structure includes three baffles and three cavities.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Turning now to a consideration of the drawing, and in particular to FIG. 1, there is shown the upper half of a gas turbine engine 10 in which the present invention can be incorporated. Within the gas turbine engine 10, air which enters the engine is compressed by the compressor 12. A portion of the high pressure air then flows into the combustor 14 wherein it is mixed with fuel and burned. The resulting expanding hot gases flow between the turbine nozzle vanes 15 and across the turbine blades 16 causing the blades and thus the turbine rotor 18 to rotate. Another portion of the high pressure air is used as cooling air to cool the combustor walls and the turbine components. That cooling air flows through the plenums 20 and 22 disposed radially inwardly and outwardly, respectively, of the combustor 14, the turbine nozzle vanes 15 and the turbine blades 16 and cools the above components in an appropriate manner.

As can best be seen in FIG. 2, the turbine nozzle vanes 15 and the turbine blades 16 are disposed within a gas flowpath 24 through which the hot gases flow after they exit the combustor 14. The gas flowpath 24 is defined by radially inner and outer boundaries. By "radial" is meant in a direction generally perpendicular to the engine centerline, designated by the dashed line 26.

The gas flowpath boundaries at the nozzle vanes 15 are defined by generally annular structures, preferably the nozzle inner and outer bands 28 and 30, respectively. The gas flowpath boundaries at the turbine blades 16 are also defined by generally annular structures, preferably by the blade platforms 32 and the shroud 34.

Because the nozzle inner and outer bands 28 and 30, the blade platforms 32 and the shroud 34 are exposed to the high temperature gases within the gas flowpath 24, they must be cooled in order to reduce structural damage, such as through thermal expansion, and to maintain satisfactory sealing characteristics. The high pressure cooling air flowing through the plenums 20 and 22 can be employed for such cooling in a manner to be described hereinafter.

The present invention comprises a multiple-impingement cooled structure such as for use in defining a boundary of a gas flowpath. The structure is configured to receive a high pressure cooling fluid, such as air, and to appropriately direct the fluid to impinge in a sequential manner upon the portions of an element of the structure which is exposed to the gas flowpath.

FIG. 3 shows the structure of the present invention employed as a shroud assembly 36 which includes as one of its elements the shroud 34. It is to be understood, however, that the present invention can also be successfully employed as a turbine nozzle band assembly or in any other appropriate manner where it is desired to cool an element exposed to high temperature.

As can be seen in FIG. 3, the structure, or shroud assembly 36, comprises an element, such as the shroud 34, including an inner surface 38 facing toward the gas flowpath 24 and an outer surface 40 facing away from the gas flowpath 24. The element, or shroud 34, also includes upstream and downstream edges 42 and 44, respectively. By "upstream" is meant in a direction from which the gases in the gas flowpath 24 flow as they approach the structure. By "downstream" is meant in a direction toward which the gases flow as they depart the structure.

The shroud 34 and shroud assembly 36 are shaped so as to properly define a boundary of the gas flowpath 24. In the case of a gas turbine engine such as that shown in FIGS. 1 and 2, the shroud 34 and the shroud assembly 36 are generally annular, more particularly the shroud 34 being generally cylindrically shaped, because the gas flowpath 24 has a generally annular shape. The shroud assembly 36 can be circumferentially continuous or it can comprise a plurality of circumferentially adjacent shroud assembly segments, in the latter case the shroud 34 being arcuate.

Again referring to FIG. 3, the element or shroud 34 includes at least one rib 46 extending from the outer surface 40 and generally parallel to the downstream edge 44. The rib 46 is preferably disposed on the shroud approximately near the center of the shroud. The function of the rib 46 will be explained hereinafter.

The structure, or shroud assembly 36, further comprises an upstream flange 48 and a downstream flange 50 disposed on opposite sides of the rib 46 and extending outwardly from the outer surface 40 of the element, or shroud 34. Preferably, the upstream and downstream flanges 48 and 50 extend from the shroud 34 at or near the upstream and downstream edges 42 and 44, respectively, thereof. When the shroud assembly 36 is generally annular, the upstream and downstream flanges extend in a generally radial direction. If necessary for enabling attachment of the shroud assembly 36 to another member, the upstream and downstream flanges 48 and 50 can include lips 52 and 54, respectively.

A first baffle 56 extends between the upstream and downstream flanges 48 and 50 and is spaced from the element, or shroud 34, and from the rib 46. A second baffle 58 extends between the downstream flange 50 and the rib 46 and is spaced between the first baffle 56 and the element, or shroud 34.

A first cavity 60 is defined within the shroud assembly 36 by the first baffle 56, the upstream and downstream flanges 48 and 50, an upstream portion of the shroud 34, the rib 46 and the second baffle 58. A second cavity 62 is defined within the shroud assembly 36 by the second baffle 58, the rib 46, the downstream flange 50, and a downstream portion of the shroud 34.

The first baffle 56 includes a plurality of impingement holes 64 through only a portion thereof for directing impingement cooling air from a source, such as the plenum 22 which is exterior to the structure, against the portion of the element, or shroud 34, within the first cavity 60. In the configuration shown in FIG. 3, the impingement cooling air flowing through the impingement holes 64 would be directed against only the upstream portion of the shroud 34.

The second baffle 58 also includes a plurality of impingement holes 66 therethrough for directing impingement cooling air from the first cavity 60 against the portion of the element, or shroud 34, within the second cavity 62. In the configuration shown in FIG. 3, the impingement cooling air flowing through the impingement holes 66 would be directed against only the downstream portion of the shroud 34.

Thus, the primary advantage of this multiple-impingement cooling arrangement over prior art single impingement cooling arrangements is that the first and second baffles 56 and 58 are arranged such that together they direct cooling air to impinge sequentially upon the portion of the element, or shroud 34, within the first cavity 60 and then upon the portion of the element within the second cavity 62. That is, the coolant flow through the first baffle 56 is concentrated such that it impinges only upon the upstream portion of the shroud 34 and then the coolant flow is concentrated again such that it impinges only upon the downstream portion of the shroud 34. In comparison, prior art single impingement cooling arrangements would disperse the equivalent coolant flow to impinge upon the entire shroud at one time. As a result, the same coolant flow through the present invention would provide greater cooling than prior art arrangements, or, less coolant flow would be required in the present invention to provide the equivalent cooling of prior art arrangements. A reduced requirement of cooling air correspondingly increases engine efficiency.

The structure, or shroud assembly 36, also comprises fluid communication means between at least one of the cavities 60 or 62 and the exterior of the structure so as to provide a means for the cooling air to exit the structure. Such fluid communication means is necessary to maintain the pressure within the cavities 60 and 62 lower than the pressure at the coolant source so that the cooling air will continue to flow into the cavities. As can be seen in FIG. 3, the fluid communication means can comprise a plurality of film cooling holes 68 through the shroud 34. Cooling air flows from the cavities 60 and 62 through the film cooling holes 68 so as to provide a film of cooling air along the inner surface 38 of the shroud. The cooling air which exits the first cav-

ity 60 through the film cooling hole 68 will thereby not be available to flow into the second cavity 62. Therefore, the number and sizes of the film cooling holes are selected such that there remains an adequate amount of cooling air to flow into the second cavity 62 to impinge upon a portion of the shroud 34 therein.

Because of the improvement in cooling of the element, or shroud 34, by the earlier described multiple-impingement cooling arrangement, film cooling of the shroud may not be required at all, or, if it is required, fewer film cooling holes 68 are required than on previous shroud configurations. Thus, mixing losses resulting from mixing of the film cooling air with the gases flowing through the gas flowpath 24 are also reduced and turbine efficiency increases.

Although the relative positions of the first and second cavities 60 and 62 within the structure, or shroud assembly 36, can be as desired, it is preferable that they be as shown in FIG. 3. The temperature of the gases flowing through the gas flowpath 24 decreases in a downstream direction as work is extracted from the gases. Thus, the upstream portion of the shroud 34 will be subjected to higher temperatures than the downstream portion. It is preferable, therefore, that the upstream portion of the shroud 34 receive the initial impingement cooling air in the first cavity 60 since the initial cooling air entering the first cavity will be cooler and of greater amount than when it enters the second cavity 62.

Refering now to FIG. 4, there is shown another embodiment of the structure of the present invention. This embodiment is similar to that shown in FIG. 3 and the same numbers are used to identify identical elements. The embodiment of the structure, or shroud assembly 70, shown in FIG. 4 comprises an element, or shroud 34, a rib 46, upstream and downstream flanges 48 and 50 and first and second baffles 56 and 58 including impingement cooling holes 64 and 66, respectively, therethrough. The structure, or shroud assembly 70, further comprises a thermal coating 72 on the inner surface 38 of the shroud 34 to improve thermal protection of the shroud. Any appropriate thermal coating can be employed, such as, for example, the thermal barrier coating described in U.S. Pat. No. 4,055,705-Stecura et al, 1977, the disclosure of which is incorporated herein by reference. Preferably, there are no film cooling holes included in this embodiment and thereby mixing losses are greatly reduced and turbine efficiency correspondingly increases.

The structure, or shroud assembly 70, includes a plurality of bleed holes 74 spaced along and extending through the downstream flange 50 so as to provide fluid communication between the second cavity 62 and the exterior of the shroud assembly 70 to permit the cooling air to exit the structure. If desired, the shroud assembly 70 can also include a plurality of bleed holes 76 spaced along and extending through the upstream flange 48 to likewise provide fluid communication between the first cavity 60 and the exterior of the shroud assembly. Although the bleed holes 74 and 76 are shown as employed in the embodiment of FIG. 4, they can also be employed in the embodiment shown in FIG. 3, either in place of or in addition to the film cooling holes 68 shown therein.

Turning now to FIG. 5, there is shown another embodiment of the structure of the present invention. This embodiment is similar to that shown in FIG. 3 and the same numbers will be used to identify identical elements. The structure, or shroud assembly 78, comprises an element, or shroud 34, and upstream and downstream flanges 48 and 50. However, rather than including only one rib, the embodiment shown in FIG. 5 includes an upstream rib 80 and a downstream rib 82 disposed between the flanges 48 and 50, each rib extending from the outer surface 40 of the element, or shroud 34. Although the spacing of the upstream and downstream ribs 80 and 82 on the shroud 34 can be as desired, it is preferable that the ribs be disposed at locations on the shroud which are approximately one third of the distance between the upstream and downstream flanges 48 and 50, such that the element, or shroud 34, is divided into three substantially equal portions.

The structure, or shroud assembly 78, comprises three baffles: a first baffle 84 extending between the upstream and downstream flanges 48 and 50 and spaced from the shroud 34 and from the upstream and downstream ribs 80 and 82, a second baffle 86 extending between the upstream rib 80 and the downstream flange 50 and spaced between the first baffle 84 and the shroud 34, and a third baffle 88 extending between the downstream rib 82 and the downstream flange 50 and spaced between the second baffle 86 and the shroud 34.

Thus, three cavities are defined within the structure, or shroud assembly 78. A first cavity 90 is defined by the first baffle 84, the upstream and downstream flanges 48 and 50, and upstream portion of the element, or shroud 34, the upstream rib 80 and the second baffle 86. A second cavity 92 is defined by the second baffle 86, the upstream rib 80, the downstream flange 50, the center portion of the shroud 34, the downstream rib 82, and the third baffle 88. A third cavity 94 is defined by the third baffle 88, the downstream rib 82, the downstream flange 50, and the downstream portion of the shroud 34.

The first, second and third baffles 84, 86 and 88 include impingement holes 96, 98 and 100, respectively, therethrough. Cooling air from a source, such as the plenum 22, is directed by the impingement holes 96 in the first baffle 84 to impinge upon the portion of the shroud 34 within the first cavity 90. That cooling air is then directed by the impingement holes 98 in the second baffle 86 to impinge upon a portion of the shroud 34 within the second cavity 92. That cooling air is then again directed by the impingement holes in the third baffle 88 to impinge upon the portion of the shroud 34 within the third cavity 94.

The structure, or shroud assembly 78, also includes fluid communication means between at least one of the cavities and the exterior of the structure to permit cooling fluid to exit the structure. Such fluid communication means can comprise the film cooling holes 68 shown in FIG. 5, or, if desired, bleed holes extending through the upstream and downstream flanges 48 and 50, similar to those shown in FIG. 4.

The cavities within the structure of any of the above-described embodiments can either be continuous around the entire structure or, when the structure is segmented, the cavities can be segmented. When the structure of the present invention comprises a generally annular shroud assembly or nozzle band assembly which comprises a plurality of circumferentially adjacent shroud assembly segments or nozzle band assembly segments, respectively, it may be preferable that the cavities, such as the first and second cavities 60 and 62 shown in FIG. 3, include an end wall 102 at each circumferential end thereof to reduce cooling air leakage between segments.

It is to be understood that this invention is not limited to the particular embodiments disclosed and it is intended to cover all modifications coming within the true spirit and scope of this invention as claimed. For example, although the embodiments of the structure of the invention have been described as including two or three baffles and cavities therein, the structure could be modified to include four or more baffles and cavities.

We claim:

1. In a shroud which defines a flowpath for hot gases in a gas turbine engine, the improvement comprising:

(a) means for directing airstreams against a first shroud portion for impingement cooling thereof and

(b) means for collecting some of the air of (a) after impingement and redirecting the collected air against a second shroud portion for impingement cooling thereof.

2. In a gas turbine engine having a shroud having an inner surface which is heated by hot gases, a method of removing heat from the inner surface, comprising the following steps:

(a) passing cooling air through a first cavity by the use of impingement cooling;

(b) transferring heat from the inner surface along a first path to cooling air within the first cavity;

(c) passing at least some of the cooling air of (b) through a second cavity;

(d) transferring heat from the inner surface along a second path to the cooling air within the second cavity by the use of impingement cooling;

wherein the first and second paths are approximately the same length.

3. A multiple-impingement cooled shroud assembly for defining the radially outer boundary of a gas flowpath and comprising a plurality of circumferentially adjacent shroud assembly segments, each of said segments comprising:

(a) an arcuate shroud including upstream and downstream edges and a rib extending radially outwardly from near the center of said shroud and parallel to said downstream edge thereof;

(b) upstream and downstream flanges extending generally radially outwardly from said shroud at near said upstream and said downstream edges, respectively, thereof;

(c) a first baffle and a second baffle, said first baffle extending between said upstream and said downstream flanges and spaced radially outwardly of said shroud, of said rib and of said second baffle for defining therewith a first cavity, said second baffle extending between said rib and said downstream flange and spaced between said first baffle and said shroud for defining therewith a second cavity, said first baffle and said second baffle each including a plurality of impingement holes therethrough for directing cooling air from a source thereof to impinge sequentially upon the portion of said shroud within said first cavity and then upon the portion of said shroud within said second cavity; and

(d) fluid communication means between at least said second cavity and the exterior of said shroud assembly.

4. The shroud assembly of claim 3 further comprising a thermal coating on the radially inner surface of said shroud.

* * * * *

United States Patent [19]
Liang
[11] Patent Number: 5,039,562
[45] Date of Patent: Aug. 13, 1991

[54] METHOD AND APPARATUS FOR COOLING HIGH TEMPERATURE CERAMIC TURBINE BLADE PORTIONS

[75] Inventor: George P. Liang, Palm City, Fla.

[73] Assignee: The United States of America as represented by the Secretary of the Air Force, Washington, D.C.

[21] Appl. No.: 527,929

[22] Filed: May 23, 1990

Related U.S. Application Data

[62] Division of Ser. No. 262,766, Oct. 20, 1988.

[51] Int. Cl.[5] B05D 3/12
[52] U.S. Cl. 427/276; 427/336; 427/355; 427/427; 415/115
[58] Field of Search 415/115; 427/276, 336, 427/355, 427

[56] References Cited

U.S. PATENT DOCUMENTS

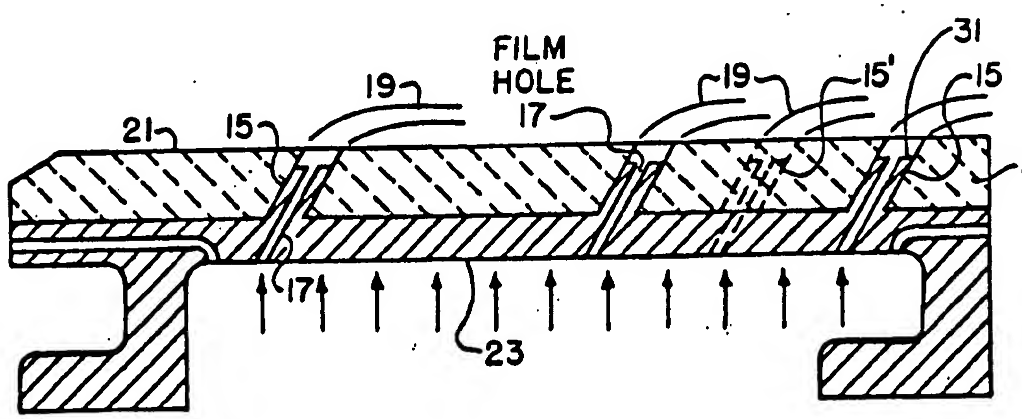
4,300,934	11/1981	DeTorre	427/276 X
4,376,004	3/1983	Bratton et al.	156/89
4,790,721	12/1913	Morris et al.	416/92 X

Primary Examiner—Shrive Beck
Assistant Examiner—Alain Bashore
Attorney, Agent, or Firm—Robert L. Nathans; Donald J. Singer

[57] ABSTRACT

An array of skewed cooling air conduits are embedded within a ceramic turbine blade for cooling an upper blade surface by producing a cool air film thereon, and by removing heat through conduction. The conduits provide a large bonding surface area in the ceramic, and are recessed to prevent clogging by the ceramic and to prevent direct contact with the hot gas stream at the upper blade surface.

12 Claims, 2 Drawing Sheets



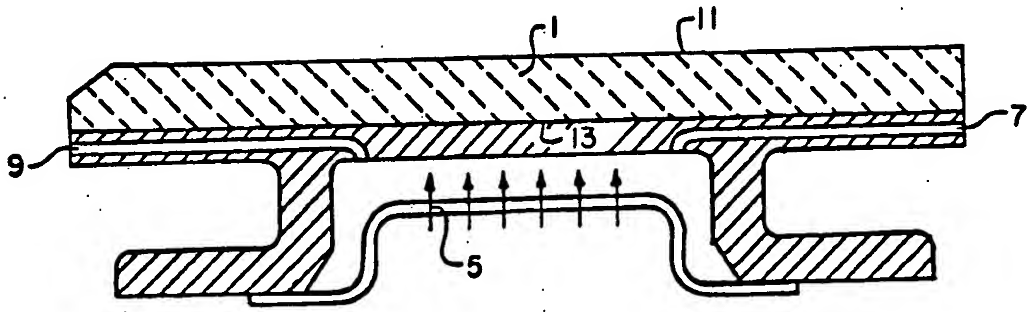


FIG. 1
PRIOR ART

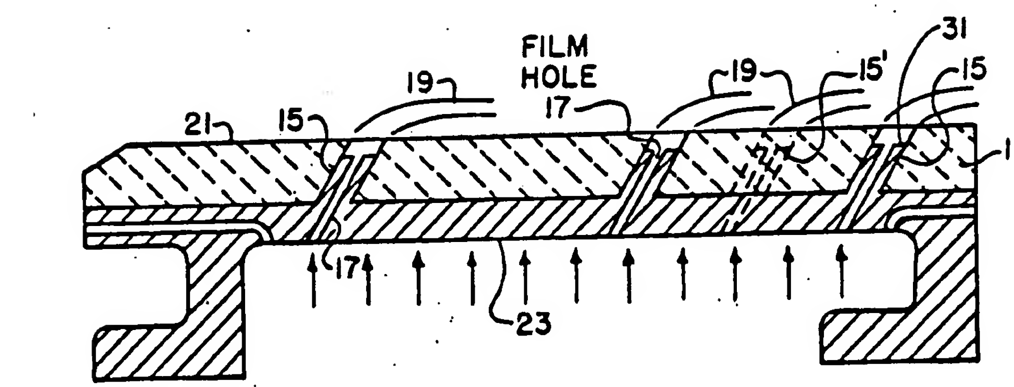


FIG. 2a

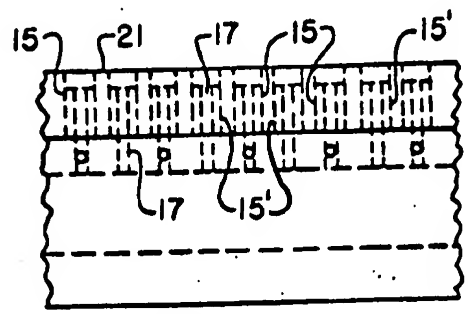


FIG. 2b

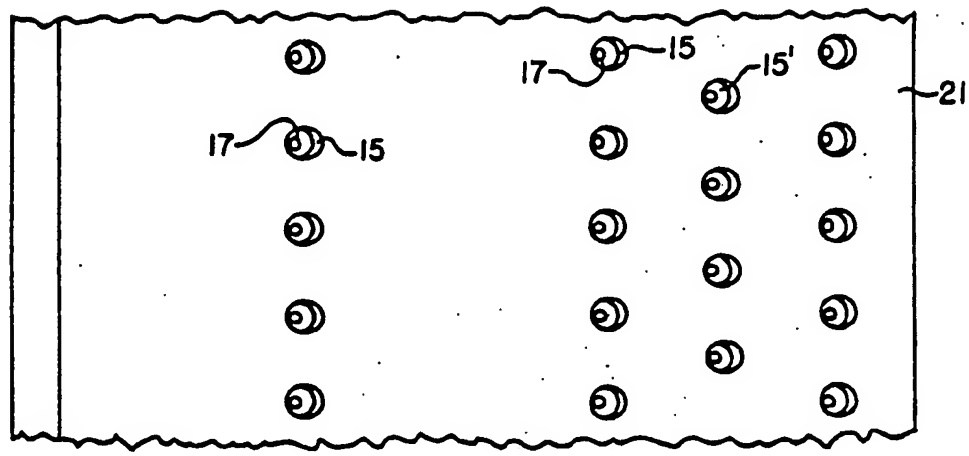


FIG. 3

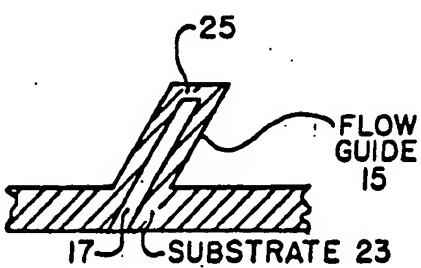


FIG. 4

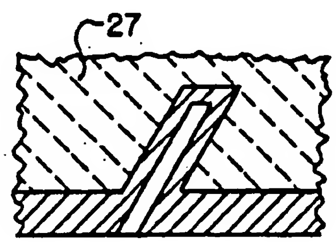


FIG. 5

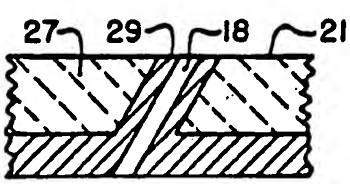


FIG. 6

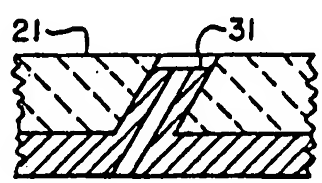


FIG. 7

METHOD AND APPARATUS FOR COOLING HIGH TEMPERATURE CERAMIC TURBINE BLADE PORTIONSSTATEMENT OF GOVERNMENT INTEREST

The invention described herein may be manufactured and used by or for the Government for governmental purposes without the payment of any royalty thereon.

This application is a division of U.S. application Ser. No. 07/262,766. filed Oct. 20, 1988.

BACKGROUND OF THE INVENTION

The present invention relates to the field of aircraft turbine blades and more particularly the cooling of ceramic blades.

Ceramic blade outer airseal life is a perennial problem in the first stage of high temperature turbines. Typical ceramic airseal failure is characterized by either cracking or partial spalling, resulting in an increased turbine tip clearance which produces a high leakage flow, reducing turbine efficiency. Such failure can be alleviated by the incorporation of a film cooling of the upper surface of the ceramic blade portion.

FIG. 1 illustrates a typical prior art ceramic airseal cooling design for a turbine tip. The upper surface 11 is the hottest portion of the ceramic body 1 and is cooled by the impingement of air illustrated by arrows 5 at the lower surface of the support substrate of the ceramic body. The cooling air flows through air ducts 7 and 9 to cool the lower surface 13 of ceramic body 1. Since the ceramic layer has a low thermal conductivity, this cooling technique produces a very hot ceramic upper surface 11 and a relatively cold lower inner surface, resulting in a high thermal gradient across the ceramic layer, to induce a high thermal stress tending to weaken the blade.

The teaching of passing cooling air through parallel conduits embedded in turbine blade portions is old in the art. The cooling air passes through, for example, honeycombed passages in U.S. Pat. 3,172,621 to Erwin, and forms a cooling air film upon the surface of the airfoil containing the cooling air honeycombed passages. U.S. Pat. No. 4,384,823 to Graham et al illustrates the concept of providing an array of cooling air tubes which transmit air to surfaces of a turbine blade. Other U.S. Pat. Nos. such as 4,684,322 and 4,249,291, illustrate the teaching of embedding cooling tubes within the body of turbine blade portions to cool surfaces thereof. It is the principle object of the present invention to provide a method of producing an array of cooling air passages embedded within a ceramic blade portion to be cooled which is believed to be more suitable than the approaches of the prior art.

SUMMARY OF THE INVENTION

In accordance with this invention, a metallic substrate supporting an array of skewed hollow projection members is provided, which convey cooling air to the upper hot ceramic blade surface to form a cooling film upon the finished blade when in use. The projection members also provide a large bonding surface for the ceramic body or blade portion, and conduct heat away from the hot upper portions of the ceramic blade. A novel method of producing this apparatus involves spraying ceramic over the projection members, thereafter machining away upper portions of the projection members to expose hollow cooling air conduits therein,

which are thereafter etched away to recess terminal cooling air conduit portions of the projection members within the ceramic body. This avoids exposing the cooling air conduits to the hot gas stream flowing over the upper blade portion, and additionally tends to reduce the possibility of the plugging of the film cooling conduits in the projection members by the ceramic material during engine operation.

For a better understanding of the present invention, together with other and further objects thereof, reference is made to the following description taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 illustrates a prior art method of cooling of the ceramic blade component;

FIG. 2 illustrates a sectional side view of a preferred product made by the aforesaid method of the invention;

FIG. 3 illustrates a plan view of FIG. 2;

FIGS. 4-7 illustrate various steps in producing the blade.

DETAILED DESCRIPTION OF THE INVENTION

FIG. 2 illustrates the preferred novel blade made by the method of the invention, which includes ceramic layer 1, bonded to substrate 3 and to tubular projections 15. The upper surface 21 is exposed to the hot gases and is cooled by cooling air passing through film hole or cooling air conduit passages 17 within projection members 15. The result is a film of cooling air 19 formed at the upper surface 21.

The plan view of FIG. 3 illustrates the array of projection members 15 and the cooling air conduit passages 17. It may be noted that staggered rows are provided at 15' to form an XY array of the film cooling air supply projection members embedded within the ceramic body 21. Thus upper portions of ceramic body at 21 are cooled by the cooling boundary layer of air 19 and are also cooled to a degree by the conduction of heat through the skewed projection members toward the lower cool substrate portion 23.

Referring now to FIGS. 4, 5, 6, and 7, an XY array of projection members 15, affixed to or formed as an integral part of substrate 23, is sprayed with ceramic material so that the tops of the projection members 25 are covered with a body of ceramic material 27. An entire array of numerous projection members 15 and 15' may be formed by molding a single part consisting of the projection members and substrate 23, staggered as illustrated in FIGS. 2 and 3. The top portions of the projection members and of ceramic body 27 are thereafter machined by, for example, grinding, until the solid terminal portions 25 of the projectors 15 are ground away to expose the hollow tubular air passageways through the projection members indicated at 18; see FIG. 6. The metallic upper portions 29 of the projectors are thereafter etched away somewhat, to form recessed terminal portions illustrated at 31 in FIG. 7, resulting in the product of FIG. 2, described above. This feature avoids exposing the conduits to the hot gaseous stream, and reduces the likelihood of the plugging of the air passageways by the ceramic material during engine operation.

Since numerous variations may be made in the practise of the invention, the scope of the invention is to be defined only by the language of the following claims and are recognized equivalents thereof.

What is claimed is;

1. Method for producing an apparatus for cooling ceramic turbine blade portions comprising the steps of:

(a) providing an array of projection members, having given lengths, affixed to a substantially planar substrate, said projection members having cooling air flow passage means therein having lengths less than the given lengths of said projection members for defining a hollow terminal tube portion adjacent a solid terminal projection member portion;

(b) covering said substrate and said projection members with a body of said ceramic material, said body having an upper surface separated from and positioned above the solid terminal portion of said projection members; and

(c) removing sufficient ceramic material and material within said solid terminal portions of said projection members to expose the cooling air flow passage means within said projection members.

2. The method of claim 1 wherein step (b) is performed by spraying said ceramic material over said substrate.

3. The method of claim 1 wherein step (c) is performed by grinding away the solid terminal portions of said projection members along with a portion of said body.

4. The method of claim 2 wherein step (c) is performed by grinding away the solid terminal portions of said projection members along with a portion of said body.

5. The method of claim 1 wherein said projection members are skewed relative to said substantially planar substrate.

6. The method of claim 2 wherein said projection members are skewed relative to said substantially planar substrate.

7. The method of claim 3 wherein said projection members are skewed relative to said substantially planar substrate.

8. The method of claim 4 wherein said projection members are skewed relative to said substantially planar substrate.

9. The method of claim 1 including the step of further removing terminal portions of said projection members for recessing the terminal portions of said projection members within said body of material.

10. The method of claim 4 including the step of further removing terminal portions of said projection members for recessing the terminal portions of said projection members within said body of material.

11. The method of claim 5 including the step of further removing terminal portions of said projection members for recessing the terminal portions of said projection members within said body of material.

12. The method of claim 8 including the step of further removing terminal portions of said projection members for recessing the terminal portions of said projection members within said body of material.

* * * * *

United States Patent [19] | [11] **4,251,185**
Karstensen | [45] **Feb. 17, 1981**

[54] **EXPANSION CONTROL RING FOR A TURBINE SHROUD ASSEMBLY**

[75] Inventor: **Karl W. Karstensen**, Peoria, Ill.

[73] Assignee: **Caterpillar Tractor Co.**, Peoria, Ill.

[21] Appl. No.: **902,016**

[22] Filed: **May 1, 1978**

[51] Int. Cl.[3] **F01D 11/08**
[52] U.S. Cl. **415/136**; 415/113; 415/175
[58] **Field of Search** 415/136, 110, 113, 116, 415/126, 134, 174, 175; 416/190, 191

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,488,867	11/1949	Judson	415/136
2,685,429	8/1954	Auyer	415/110
2,962,256	11/1960	Bishop	415/136
3,141,651	7/1964	Moyer	415/136
3,391,904	7/1968	Albert et al.	415/116
3,807,891	4/1974	McDow et al.	415/138
3,959,966	6/1976	Pearce et al.	415/139
3,966,356	6/1976	Irwin	415/113
3,975,901	8/1976	Hallinger et al.	415/115
3,986,720	10/1976	Knudsen et al.	415/136
3,990,807	11/1976	Sifford	415/136
4,023,731	5/1977	Patterson	415/116
4,023,919	5/1977	Patterson	415/134
4,050,843	9/1977	Needham et al.	415/116
4,157,232	6/1979	Bobo et al.	415/116

FOREIGN PATENT DOCUMENTS

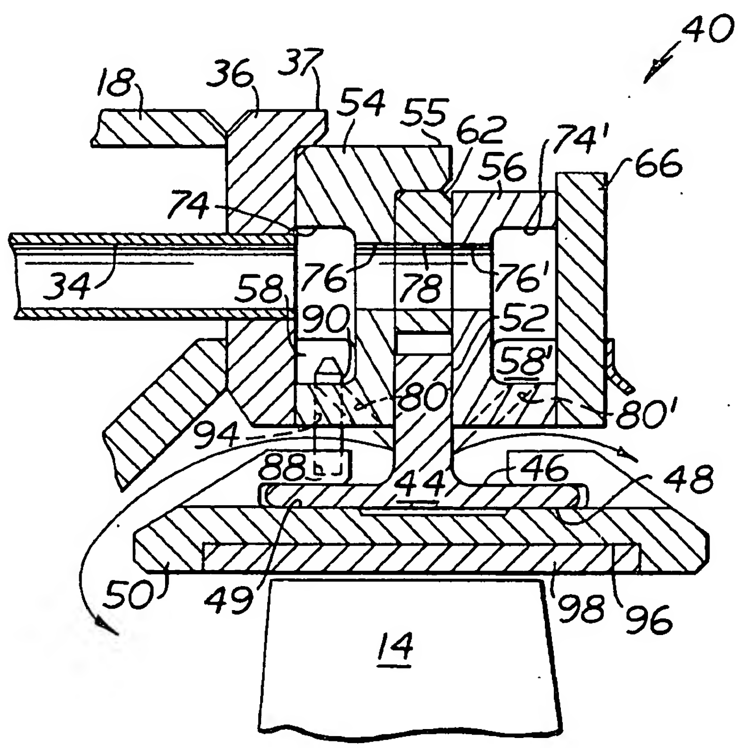
2422533	12/1974	Fed. Rep. of Germany	415/136
790854	2/1958	United Kingdom	415/136

Primary Examiner—Everette A. Powell, Jr.
Assistant Examiner—A. N. Trausch, III
Attorney, Agent, or Firm—Phillips, Moore, Weissenberger, Lempio & Majestic

[57] **ABSTRACT**

A turbine shroud assembly includes an expansion control ring to support segmented rotor shrouds. The expansion control ring is restrained by adjacent manifold rings, yet free to thermally expand radially outwardly without loss of axial alignment with the associated turbine wheel. The ported manifold rings are positioned on either side of an outwardly extending leg of the expansion control ring to direct cooling fluid delivered thereto toward the expansion control ring. A spacer ring surrounds the expansion control ring and restrains the expansion control ring relative to the manifold rings. The spacer ring maintains axial alignment of the expansion control ring with the turbine wheel. Cooling fluid is exhausted into the main hot gas stream, both upstream and downstream of the turbine wheel thus substantially preventing hot gases from affecting the expansion control ring.

9 Claims, 5 Drawing Figures



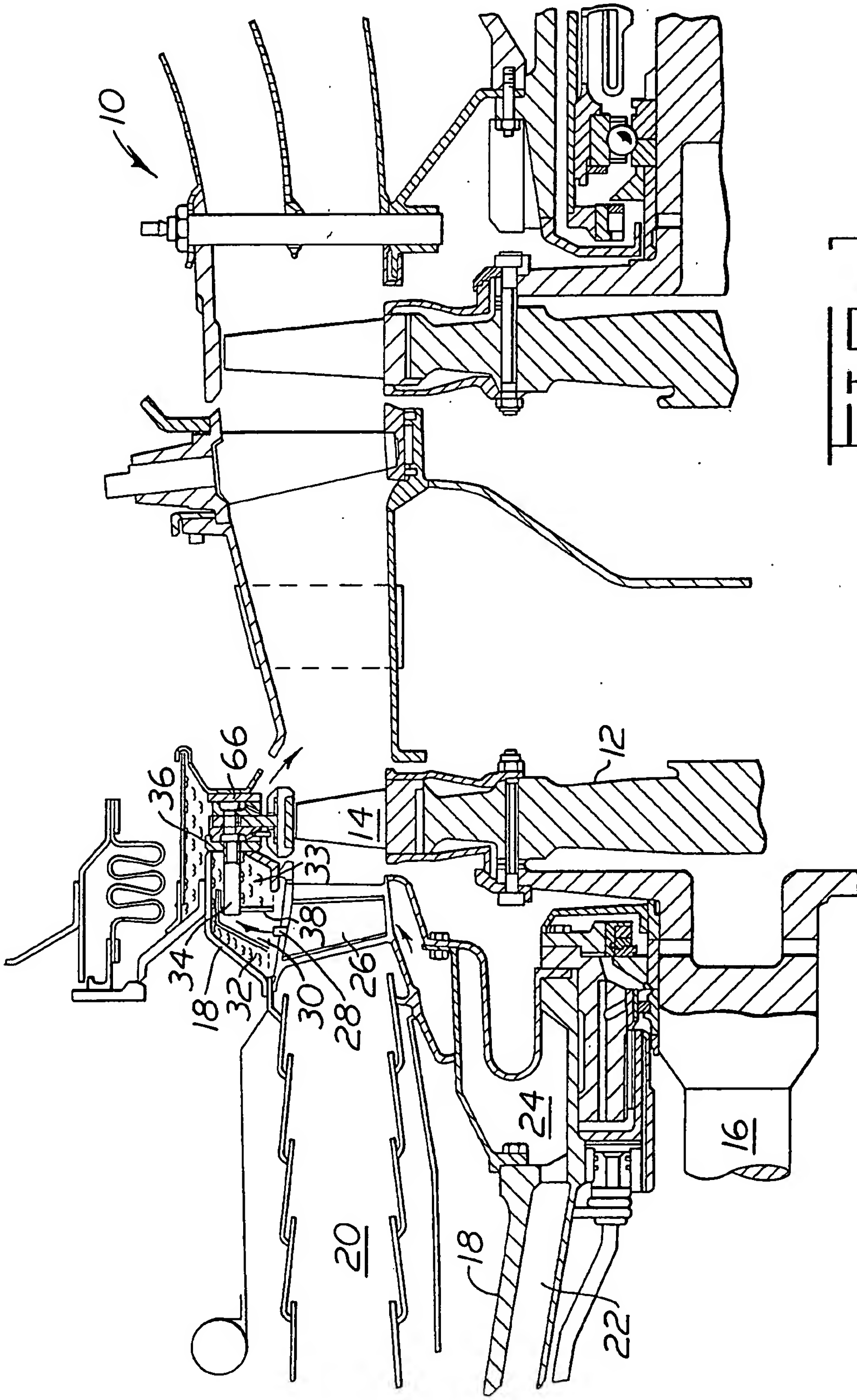


FIG. 1

Fig. 2

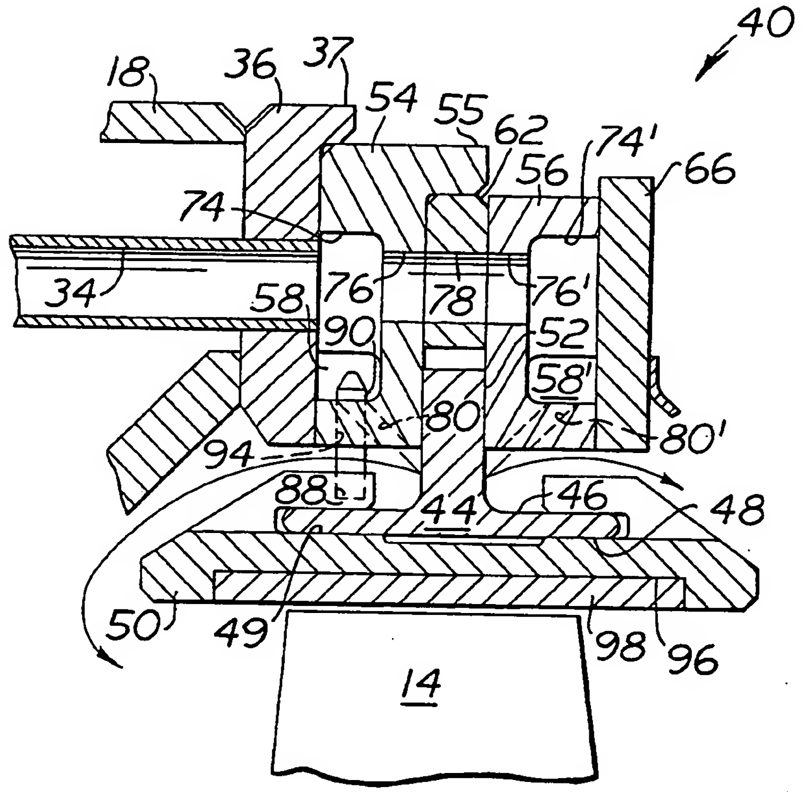

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Fig. 3

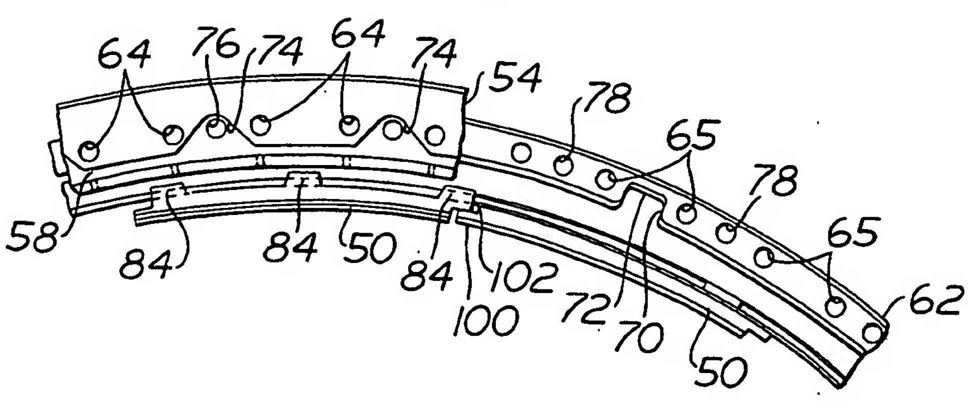
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FIG. 4

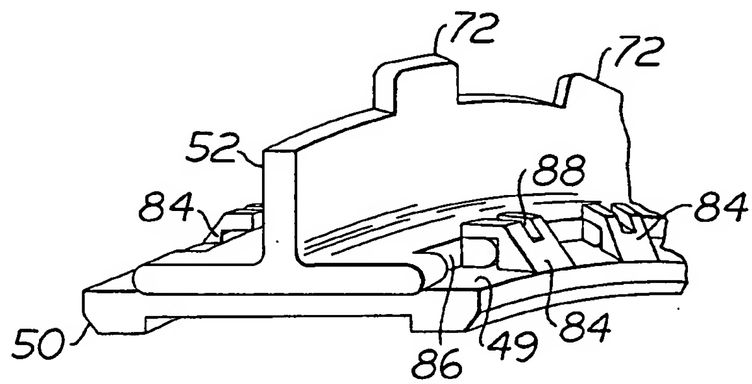
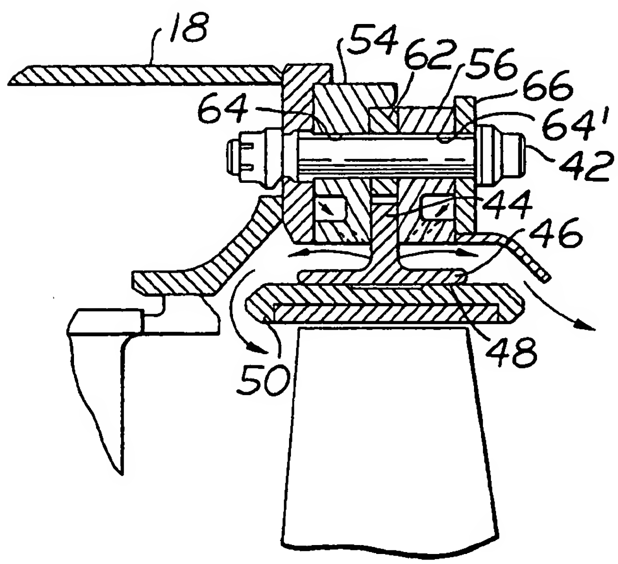


FIG. 5



EXPANSION CONTROL RING FOR A TURBINE SHROUD ASSEMBLY

BACKGROUND OF THE INVENTION

This invention relates to a shroud assembly for a turbine engine. In particular, it relates to cooling of a shroud assembly in a gas turbine engine.

Cooling of the shroud surrounding the turbine wheel in a turbine, presents rather unique problems. The shroud surrounding the turbine blades must be in close proximity to the blades in order to maintain efficiency in the turbine engine. Notwithstanding temperature, the turbine wheel must rotate freely, both at start up and during operation. It is a characteristic of turbine engines that the turbine wheel operates at a relatively high temperature. Similarly the shrouds surrounding the turbine wheel operate at a relatively high temperature. It usually is a characteristic of turbines that the material from which the turbine wheel and turbine wheel blades are made will expand at a differing rate than the surrounding shroud structure. Although it would be possible to make the turbine wheel and blades of the same material as the shroud structure, heating of the two elements of the turbine (i.e. the wheel/blades and the shroud) will differ. Experience shows that the shroud usually operates at a higher temperature than the turbine wheel. Therefore, if the two elements are made of the same material, the shroud will expand at a faster rate and finally obtain a relatively larger inside diameter than the outside diameter of the steady state expanded turbine wheel. In this situation, some of the hot fluid powering the turbine wheel will bypass the turbine blades and further may cause unnecessary turbulence in the vicinity of the turbine blades, thus resulting in excessive fuel consumption.

Accordingly, it is desireable to provide cooling in the vicinity of the turbine shroud to decrease the difference in expansion between the turbine shroud and the turbine wheel and blades. Cooling fluid for turbine engines of the gas turbine type used in propelling aircraft is readily available either from atmospheric air flow over an uninsulated engine case, or bleed air from the compressor, or, in the case of a turbofan engine, air bled from the fan.

In an industrial gas turbine engine of the embodiment of this invention, atmospheric air or fan bleed air is not available. Furthermore, the engine case in an industrial gas turbine engine is generally heavily insulated to prevent heat loss within the engine itself, thus ambient air is of little use. Compressed air flow from the compressor stage of an industrial gas turbine engine is usually communicated directly to a heat exchanger. The heat exchanger serves two purposes, first to warm the incoming air for subsequent combustion in the gas turbine itself and secondly, to cool the exhaust gases before discharge into the atmosphere. It is impractical to utilize the compressed air from the heat exchanger with its recuperated heat for cooling of the turbine shroud since the temperature of this air is excessive. On the other hand, air may be bled directly from the compressor stage and communicated through appropriate manifolding to cool the various gasifier turbine parts. This compressor bleed air has a relatively cool temperature established primarily by the compression ratio, and secondarily by conduction from the hot engine case.

Use of bleed air from the turbine compressor should be limited in order to achieve the highest degree of engine efficiency. In earlier industrial gas turbine engines, air was supplied in a random manner to the shroud structure surrounding the turbine blades. Furthermore, the material utilized in earlier shroud structures was usually chosen primarily for its strength with secondary attention paid the coefficient of thermal expansion.

Furthermore, in earlier industrial gas turbine engines wherein the cooling air was provided in an haphazard fashion to the shroud rings, no attempt was made to isolate the cooling air from the surrounding hot structure thus; by the time the cooling air arrived at the shroud structure, a good deal of its cooling potential had been lost due to temperature increases through contact with hot surfaces of the gas turbine engine. Finally, the large plenum chamber arrangement involved in earlier gas turbine engines resulted in a drop in pressure of the cooling air to the extent that hot gas was able to enter the plenum chamber and further degrade the cooling.

Attempts to maintain a smooth gas flow through the turbine and past the turbine wheel, resulted in the turbine shroud essentially being made an integral part of the turbine casing. Thus, the adjacent relatively high temperatures of the turbine casing were conducted to the turbine shroud with a concomitant expansion of the turbine shroud. Even though attempts have been made to restrain the expansion of the turbine shroud, efforts along this line have not been entirely successful. To compound the problem, reduction in a diameter or maintenance of the same diameter of the turbine shroud ring resulting from impingement of cooling air thereon was resisted by the mechanical restraints imposed on the turbine shroud ring by expansion of the hotter portions of the turbine engine casing.

Finally, earlier gas turbine engines, both of the industrial type and the aircraft type, usually used an overlapping segmented shroud assembly to permit thermal expansion within each segment while not substantially increasing the inside diameter of the entire shroud structure. These individual shroud segments were mounted in various ways with cooling air usually directed toward them in the aforedescribed haphazard manner. A possible disadvantage resulted from such a structure. A true circular opening for the turbine wheel was difficult to achieve because of the segmented nature of the shroud assemblies themselves. Therefore, the clearance between the turbine shroud assembly and a turbine blade had to be adjusted to account for a possible out of round condition. This adjustment resulted in an unnecessary loss of efficiency.

SUMMARY OF THE INVENTION

The present invention is directed to overcoming one or more of the problems as set forth above.

Broadly stated, the present invention is a turbine shroud assembly comprising an expansion control ring defining an inner cylindrical surface. Manifolds are provided for directing a cooling fluid toward preselected locations on the expansion control ring. A spacer ring axially associates the expansion control ring with the manifolds.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a sectional view of a portion of a gas turbine engine in which the shroud assembly described herein may be used.

FIG. 2 is a sectional view in greater detail of the shroud assembly shown in FIG. 1 and described herein.

FIG. 3 is a partial elevation view of the shroud assembly shown in FIG. 2 with a portion broken away to illustrate the structure of the expansion control ring.

FIG. 4 is a perspective view of a portion of a rotor shroud segment shown positioned on a portion of the expansion control ring.

FIG. 5 is a sectional view of the turbine shroud assembly showing one of the bolt members fixing the assembly to the turbine case.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

A portion of a gas turbine engine 10 is shown in FIG. 1. Gas turbine engine 10 includes a gasifier turbine wheel 12 upon which a plurality of turbine vanes 14 are mounted. Turbine wheel 12 is fixed to a shaft 16 which is mounted for rotation in a turbine case 18 in which a combustion chamber 20 is also affixed. Shaft 16 rotates a compressor, (not shown) from which a quantity of cooling fluid or air is bled off to a passage 22 and communicated to plenum chamber 24 for subsequent communication to the interior of a plurality of turbine nozzle vanes 26. The cooling arrangement described thus far is more thoroughly set forth in U.S. patent application Ser. No. 729,746 now U.S. Pat. No. 4,086,757 assigned to the assignee of this invention.

Cooling fluid communicated to turbine nozzle 26 is relieved through a vent 28 into a chamber 30 which is annular in form. Chamber 30 is surrounded by insulating material 32 which may be of any material well known in the art, such as ceramic fiber. Communicating with chamber 30 are a series of tubes 34 fixed to a flange 36 which surrounds turbine wheel 12. Flange 36 is fixed to turbine case 18 and provides a portion of the support for the turbine shroud assembly 40 (see also FIGS. 2 and 5).

Affixed to the other opposite end of tubes 34 is a sheet metal member 38 which forms an annulus. Insulating material 33 is positioned in the annulus and around tubes 34 to minimize the temperature rise in the cooling fluid as it is communicated from the nozzle vane liner 26 to the turbine shroud assembly.

The turbine shroud assembly 40 is affixed to flange 36 by a plurality of bolt members 42 in the manner shown in FIG. 5. The shroud assembly 40 is comprised of an expansion control ring 44 which has a generally T shaped cross sectional configuration. The expansion control ring 44 defines a cross bar portion 46 which in turn has an inner cylindrical surface 48 to which a plurality of rotor segments 50 mountingly abut.

Extending radially outwardly from cross bar portion 46 is a leg portion 52. Positioned on opposed sides of leg portion 52 is means for forming a manifold to communicate cooling air to the intersection of leg portion 52 and cross bar portion 46. This means is comprised of first and second manifold rings 54 and 56 respectively. Manifold rings 54 and 56 are similar with structure differing on the outer perimeter thereof as indicated in FIG. 2. First and second manifold rings 54 and 56 have interposed there between and positioned radially outwardly of expansion control ring 44, a spacer ring 62 which has a width slightly greater than leg portion 52 of expansion control ring 44. Spacer ring 62 radially associates expansion control ring 44 with the manifold means.

Referring now to FIG. 3 in conjunction with FIG. 5, it can be seen that the manifold ring 54 and manifold ring 56 are formed with a plurality of fastening holes 64 and 64' respectively. Similarly, spacer ring 62 is formed with a plurality of fastener holes 65. The plurality of bolt member 42 previously mentioned in relation to FIG. 5, are passed through these fastener holes to flange 36 and a flange 66 also affixed to turbine housing 18. It should be noted that flange 36 is formed with a rearwardly extending lip 37 at its outer periphery and overlapping manifold ring 54. The manifold ring 54 is also formed with a rearwardly extending lip 55 overlapping spacer ring 62.

The spacer ring 62 is formed with a plurality of parallel sided notches 70 adapted to receive lugs 72 formed on the outer perimeter of leg portion 52 of expansion control ring 44. Referring to FIG. 3, it can be seen that each lug 72 is mated with a corresponding notch 70 with expansion room provided between lug 72 and notch 70. Axial alignment of expansion control ring 44 with turbine wheel 12 is not affected during thermal expansion of expansion control ring 44 because the parallel sides of the notches 70 and the corresponding lugs 72 require uniform expansion of the ring. Therefore, the expansion control ring 44 may expand to a different degree from the turbine casing itself, without affecting the concentricity of expansion control ring 44.

Each manifold ring, 54 and 56, is formed with a plurality of relieved areas 74 and 74' having a generally triangular shape as indicated in FIG. 3, although other shapes would serve adequately. Each relieved area 74 and 74' communicates its widest part with the corresponding groove 58 in manifold ring 54 and the groove 58' in the manifold ring 56. A bore 76 (see FIG. 3) is formed generally at the apex of the triangular shaped relieved area 74. The bore 76 communicates with a bore 78 formed in spacer ring 62 which in turn, communicates with a corresponding bore 76' in the second manifold ring 56 as indicated in FIG. 2. This second bore 76' in turn communicates with the corresponding relieved area 74' of second manifold ring 56. A plurality of orifices or ports 80 communicate groove 58 (see FIG. 2) with the area adjacent expansion control ring 44. Specifically, each port 80 in the first manifold ring 54 is directed toward one side of the expansion control ring 44 in the vicinity of the intersection of the leg portion 52 and the cross bar portion 46. A similar plurality of ports 80' is formed in manifold ring 56, thus cooling fluid communicated through the bores 76, 78, and 76' to relieved area 74' is controlably directed towards the opposite side of expansion control ring 44 in the specific vicinity of the intersection of leg portion 52 and a cross bar portion 46.

The particular structure described to this point provides cooling to expansion control ring 44 which may be formed of a particular low expansion alloy such as "Hastelloy Alloy's". This particular material has been found to have sufficient high temperature strength for this application while retaining a relatively low co-efficient of thermal expansion. The relatively loose contact between expansion control ring 44 and the adjacent manifold rings and spacer, creates a relatively high resistance to heat conduction from adjacent engine parts which could obviate efforts to cool expansion control ring 44. The lug and notch connection between expansion control ring 44 and spacer ring 62 eliminate mechanical stresses between these two parts which would tend to resist the reduction in diameter of expansion control ring 44 resulting from cooling air applied to

the intersection of leg portion 52 and cross bar portion 46.

Referring to FIG. 4, a perspective view of a rotor shroud segment 50 is shown mounted on a portion of expansion control ring 44. Each rotor shroud segment 50 is formed with a plurality of inwardly facing tabs 84 which are formed on the outer surface 49 thereof, to overlap cross bar portion 46 of expansion control ring 44. The expansion control ring 44 has a plurality of loading notches 86 spaced at a distance substantially equal to the distance separating inwardly facing tabs 84, thus the plurality of rotor shroud segments 50 can be placed on expansion control ring 44 by orienting tabs 84 in notches 86 and then sliding the plurality of rotor shroud segments to the position indicated in FIG. 4. It will be noted that one of the two center tabs 84 has formed therein a notch 88 which forms part of a socket for a dowel 90 to be positioned in. A corresponding bore 94 is formed in manifold ring 54. Thus, it can be seen in FIG. 2 that the dowel 90 is circumferentially orients each individual rotor shroud segment 50 on expansion control ring 44. The rotor shroud segments 50 can also be made of the same low expansion alloy as expansion control ring 44. Additionally, the outer surface 49 which contacts or mates with the inner cylindrical surface 48 preferably has substantially the same radius of curvature as the mating cylindrical surface 48.

It has been found that the central location of the dowel pin 90 permits expansion of each individual rotor shroud segment 50 without affecting the expansion of the next adjacent rotor shroud segment. That is, referring to FIG. 3, each rotor shroud segment 50 expands outwardly from the center relative to expansion control ring 44 rather than expanding from a locking point at one end.

Referring to FIG. 2, a cross section of rotor shroud 50 is shown in relation to the associated turbine blade vane 14. It can be seen that rotor shroud segment 50 is formed with a longitudinal groove 96. Fixed in longitudinal groove 96 is an abradable material 98 in the manner well known in the art. This abradable material acts to protect the tip of turbine vane 14 in the event turbine vane 14 contacts the rotor shroud segment.

Referring now to FIG. 3, it can be seen that the ends of each rotor shroud segment 50 are formed to overlap one another. That is, the first end 100 overlaps the second end 102 of the next adjacent rotor shroud segment.

Referring now to FIG. 1 for a better understanding of the operation of this invention, it can be seen that cooling air is provided from the compressor portion of the turbine engine to passages formed in nozzle vanes 26. After cooling nozzles 26, the cooling air passes outwardly of each nozzle through vent 28 into chamber 30 and thence into tubes 34. Each tube 34 is insulated by material 33 such as a ceramic fiber material. This material prevents an increase of heat in the cooling air passing from nozzle vanes 26 en-route to shroud assembly 40. Air is communicated to manifold ring 54 from tubes 34 at the plurality of relieved areas 74. Concurrently, a portion of the cooling air passes from the relieved area 74, through the bores 76,78 and 76' and to relieved area 74'. The cooling air in the relieved areas 74 and 74' passes through ports 80 and 80' and is controlably directed against the intersection of leg portion 52 and cross bar portion 46 of expansion control ring 44.

As can be seen by heavy arrows in FIG. 2, cooling air passes outwardly between the rotor shroud segments 50 and the turbine casing 18 and into the main hot gas stream at locations upstream and downstream of turbine vane 14. This air flow path is particularly advantageous in that hot gases in the turbine mainstream are effectively prevented from reaching the expansion control ring.

It should be noted that the rotor shroud segments 50 are not in direct contact with the adjacent portion of the turbine casing, thus heat is not effectively conducted from the casing directly to the rotor shroud segments. The connection of each rotor shroud segment 50 to turbine casing 18 is through expansion control ring 44 and in particular, through leg portion 52. Since cooling air is controlably directed against the leg portion 52, conduction of heat from the turbine casing 18 through the leg portion 52 is lessened while the leg portion and the cross bar portion themselves are cooled by the air impinging thereupon.

Although this invention has been described in relation to a particular embodiment, it is not to be considered so limited. The invention is to be considered limited only by the appended claims.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

1. A turbine shroud assembly comprising:

an expansion control ring defining an inner cylindrical surface;

manifold means for directing a cooling fluid toward preselected locations on said expansion control ring and means for coaxially supporting said expansion control ring relative to said manifold means under varying temperature conditions;

said expansion control ring defining a generally T-shaped cross-section, including a leg and a cross-bar portion, the upper surface of the cross-bar portion of the T forming the inner cylindrical surface, the leg extending radially outwardly from said cross-bar portion;

said manifold means comprising first and second manifold rings, said first and second manifold rings being positioned on opposite sides of the leg portion of the expansion control ring and each of said first and second manifold rings defining a plurality of fluid ports, said ports being oriented toward opposite sides of the leg portion of said expansion control ring.

2. The turbine shroud assembly of claim 1 wherein the means for coaxially supporting the expansion control ring comprises a spacer ring.

3. The turbine shroud assembly of claim 1 wherein the supporting means comprises a spacer ring said spacer ring having an axial thickness slightly greater than the leg portion, said spacer ring being interposed between the first and second manifold rings and radially outside said leg portion.

4. The turbine shroud assembly of claim 3 wherein the expansion control ring defines an axis and a plurality of projecting lugs extending outwardly from the leg portion thereof at predetermined locations, and wherein the spacer ring defines a plurality of notches formed in the inner perimeter thereof, said notches each having a width substantially equal to the width of said lugs and being positioned for receiving a respective lug.

5. The turbine shroud assembly of claim 1 wherein the first and second manifold rings and the spacer ring define a plurality of aligned holes,

and each of said first and second manifold rings define one and the other substantially parallel flat surfaces

said one flat surface further defining a circumferential groove, and said one flat surface further defining a plurality of relieved areas communicating a predetermined number of the aligned holes of each of said first and second manifold rings with said circumferential groove.

6. The turbine shroud assembly of claim 5 wherein the manifold means comprises first and second manifold rings, said first and second manifold rings being positioned on opposite sides of the leg portion of the expansion control ring.

7. The turbine shroud assembly of claim 6 wherein the first and second manifold rings define a plurality of fluid ports, said ports being oriented toward opposite sides of the leg portion of said expansion control ring.

8. The turbine shroud assembly of claim 7 wherein each of the ports communicates the groove with the inner perimeter of respective first or second manifold rings, said ports being directed at angles sufficient for impinging fluid from said ports generally on the intersection of the leg portion and the cross bar portion of the expansion control rings.

9. The turbine shroud assembly of claim 1 wherein the shroud assembly is comprised of a plurality of segments circumferentially mounted on said inner cylindrical surface of said expansion control ring.

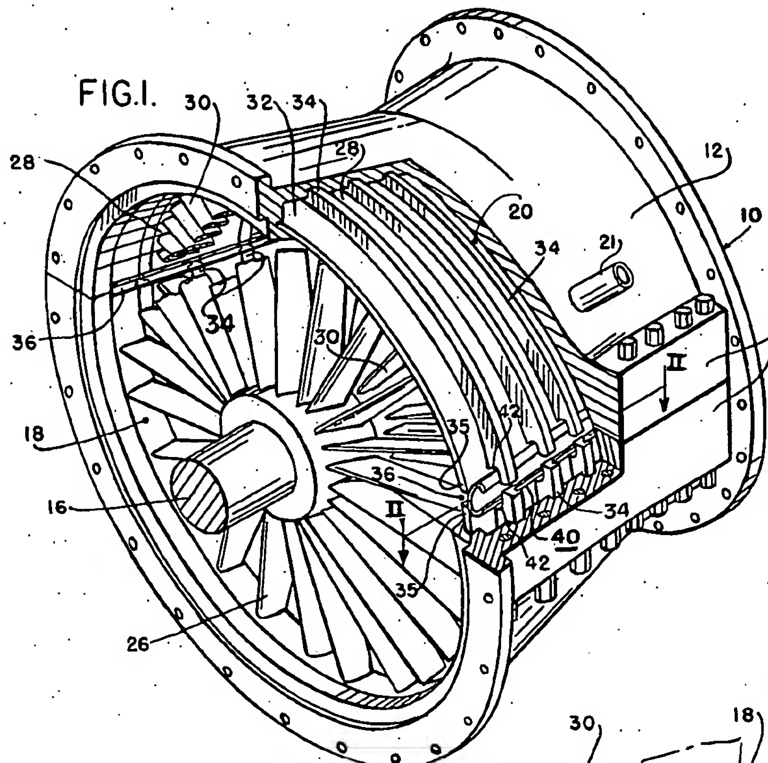
* * * * *

July 4, 1961 R. L. TASSONI 2,991,045

SEALING ARRANGEMENT FOR A DIVIDED TUBULAR CASING

Filed July 10, 1958 2 Sheets-Sheet 1

FIG.1.



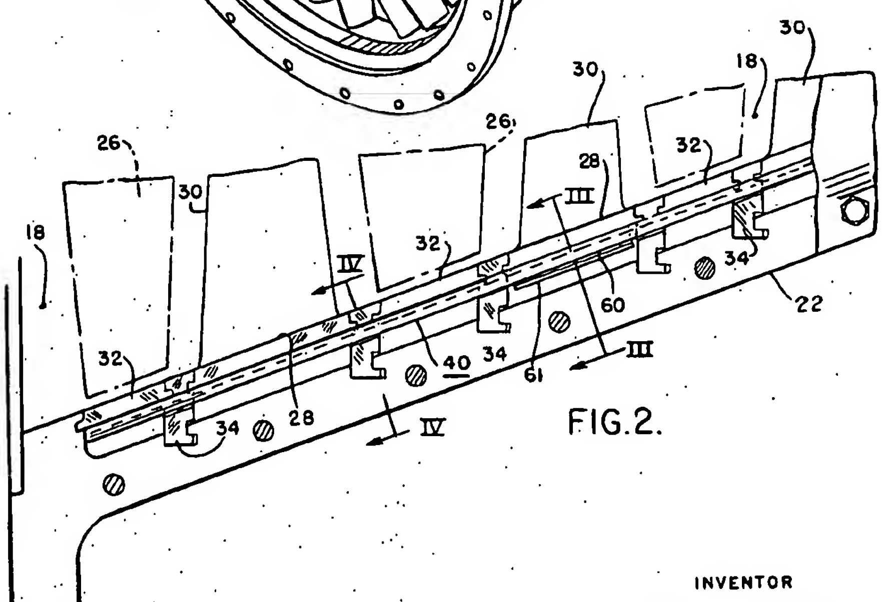


FIG.2.

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FIG.3.

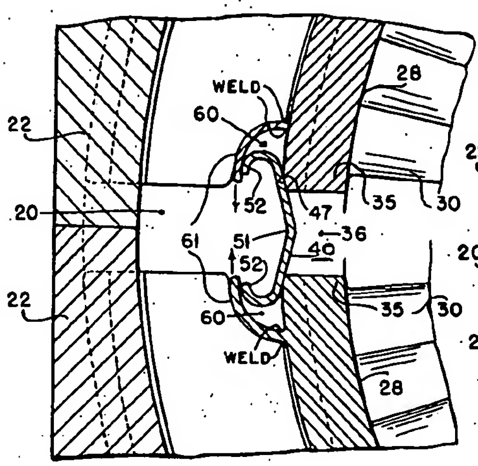
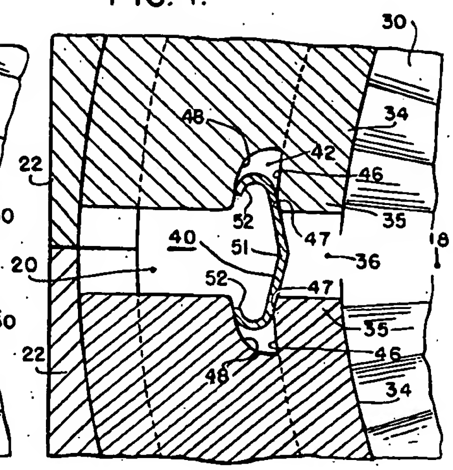


FIG.4.



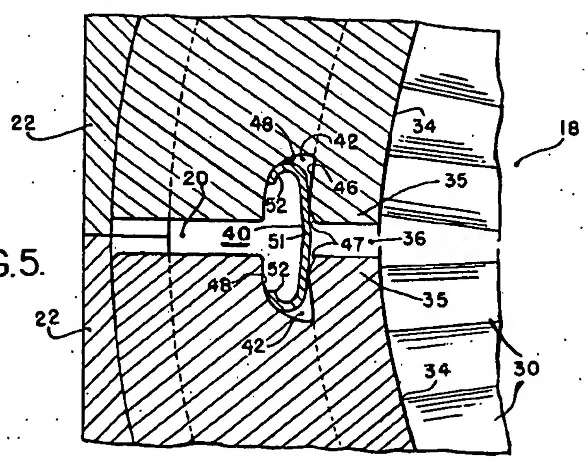


FIG.5.

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SEALING ARRANGEMENT FOR A DIVIDED TUBULAR CASING

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6 Claims. (Cl. 253—39)

This invention relates to a sealing device for a casing and has for an object to provide an improved sealing device for restricting fluid flow through a space that varies in size as the casing expands and contracts due to heating and cooling thereof, respectively.

In certain gas turbines inner and outer tubular casings are utilized, the inner casing comprising a plurality of pairs of axially aligned semi-circular rings supported by the outer casing and disposed with their ends in spaced juxtaposition. Thus, a pair of diametrically opposed elongated spaces are provided to accommodate expansion of the rings. Previously, the practice has been to provide a separate seal member at the part of the space formed by the adjacent ends of each pair of rings. It is another object of the present invention to provide a multiple continuous seal arrangement for each of the elongated spaces formed by all of these rings.

During operation, the inner casings of gas turbines are subjected, in many instances to a gas at a high temperature and high pressure, causing the inner casing to expand a substantial amount. However, when the gas turbine is removed from service, it must be free to contract upon cooling. Therefore, it is another object of this invention to provide a seal that will restrict fluid leakage through the aforementioned space and accommodate this thermally induced expansion and contraction.

The present invention has been incorporated in a gas turbine including an inner casing and an outer casing of tubular shape. The inner casing defines a flow path for high temperature and high pressure gases and comprises a plurality of pairs of approximately semi-circular rings, interlocked one to the other and supported in a manner permitting their thermally induced expansion and contraction. The rings include terminal portions that define a pair of diametrically opposed horizontal spaces. Upon expansion the juxtaposed terminal portions move toward each other, thereby reducing the width of the spaces.

A longitudinally extending resilient seal member of generally C-shaped cross section is provided, having a sealing surface portion in abutment with the terminal portions of the rings and straddling the space formed therebetween for restricting fluid flow therethrough. The transverse marginal portions are slidably received in grooved wall members secured to some of the rings and bias the sealing surface portion against the terminal portions of the rings.

Upon heating, the terminal portions expand and move closer to each other, thereby tending to move the sealing surface portions laterally toward the walls, but such lateral movement of the sealing surfaces is restricted by the walls. Thus, although the cross-sectional shape of the seal member tends to deform, the sealing surface portions remain in contact with the terminal portions of the rings. The grooves in the walls, however, permit the seal member to move in the longitudinal axial direction.

Upon cooling, the terminal portions contract and move farther apart from each other. Since the seal member is resilient, it tends to return to its original cross-sectional shape, thereby maintaining its sealing properties.

The foregoing and other objects are effected by the invention as will be apparent from the following description and claims taken in connection with the accompanying drawings, forming a part of this application, in which:

FIG. 1 is a perspective view of a gas turbine in which a part of the outer casing has been broken away to show the present invention;

FIG. 2 is a view taken along the line II—II of FIG. 1 and looking in the direction indicated by the arrows;

FIGS. 3 and 4 are sectional views, taken along the lines III—III and IV—IV of FIG. 2, respectively, and looking in the direction indicated by the arrows, showing the seal member in the cold position; and

FIG. 5 is a sectional view similar to FIG. 4 but showing the seal member in the hot position.

Referring to FIG. 1, a gas turbine 10 is illustrated comprising an outer tubular casing 12 and an inner tubular casing 14, the latter defining in conjunction with a turbine rotor 16 a motive fluid flow path 18. The casings 12 and 14 jointly define an annular chamber 20 to which is supplied, through a conduit 21, a pressurized cooling fluid from a suitable source, not illustrated, for cooling the outer casing 12 and the inner casing 14. The inner casing 14 and the outer casing 12 are each circular in cross section and are divided into a top half and a bottom half. Each half of the outer casing is provided with flanges 22 connected by suitable bolting.

The inner casing 14 encompasses a plurality of rows of rotating blades 26 supported by the rotor 16 and comprises a plurality of shroud rings 28 to which are secured stationary blades 30. Bridging rings 32 extend between the shroud rings 28 and both are supported by the outer casing 12 by being keyed to opposite sides of radially extending support rings 34 that are, in turn, keyed to the outer casing.

The shroud rings, bridging rings and support rings are divided into pairs of upper and lower semi-circular segments or half rings and their juxtaposed terminal portions 35 jointly define a pair of longitudinally extending spaces 36 disposed diametrically opposite each other. Fluid communication between the flow path 18 and the chamber 20 through the spaces 36 is restricted by seal members 40 constructed in accordance with the present invention and one of which will be hereafter described.

As best illustrated in FIGS. 4 and 5, the seal member 40 is approximately C-shape in cross section and supported and biased into sealing position by grooves 42 formed in the supporting rings 34. The grooves or recesses 42 are formed by an inner arcuate wall 46 having a shape conforming to that of the adjacent shroud ring and bridging ring so that a part of the wall 46 forms, with portions of the outer walls of the shroud rings and bridging rings adjacent the space 36, continuous and longitudinally extending surfaces 47 above and below the space 36. Further, the grooves 42 are formed by an outer arcuate or approximately quarter round wall 48.

The seal member 40 is formed of flexible and resilient material of uniform thickness and comprises a longitudinally extending continuous central sealing surface portion 51 of curved shape, for example the V-shape illustrated, portions of which abut the continuous surfaces 47 at all times. The seal member 40 further includes opposed transverse marginal portions 52 of greater curvature extending from the sealing surface. The marginal portions 52 are biased by the outer wall 48 toward the continuous surfaces 47.

The seal member 40 and the grooves 42 are proportioned and arranged so that in the cold position of the inner casing, when the terminal portions 35 are farthest apart, as illustrated in FIG. 4, the central portion 51 attains its maximum curvature and the marginal portions 52 are closest to each other. In the cold position, the central portion extends substantially to the right as viewed in FIG. 4, of a vertical plane passing through the upper

and lower continuous surfaces 47. Upon heating of the inner casing, and subsequent expansion incident thereto, the terminal portions move closer to each other and the continuous surfaces 47 slide along the central seal portion tending to move the seal member to the left, as viewed in FIG. 4. However, such movement is opposed by the walls 48, causing the seal member to deform and elongate in vertical direction. The grooves 42 are large enough in the vertical direction to allow the marginal portions 52 to move along the wall 48 with a camming action, thereby maintaining the central seal portion 51 in biased abutment with the continuous surfaces 47 until, as illustrated in FIG. 5, the hot position is attained.

Viewed in another light, during movement from the cold position to the hot position, the restraint upon movement to the left imposed upon the marginal portions 52 of the seal member by the walls 48 tends to pivot the former in opposite directions about longitudinal axes coincident with the surfaces at which the central sealing surfaces 51 contact the continuous surfaces 47. Thus, as viewed in FIG. 4, the upper marginal portion tends to rotate clockwise about the surface of contact between the sealing surface 51 and the upper continuous surface 47. Similarly, the lower marginal portion tends to rotate counterclockwise about the surface of contact between the sealing surface 51 and the lower continuous surface 47.

Upon cooling of the inner casing, the terminal portions 35 will tend to move farther apart, causing the continuous surfaces 47 and recesses 42 to move away from each other. However, the central sealing surface 51 is maintained in biased abutment therewith as the flexible seal member returns to its original cross-sectional shape, illustrated in FIG. 4.

It will be noted that the seal member and grooves are symmetrical about a horizontal plane. This symmetry tends to maintain the seal centered in its proper position and in sealing relation when the inner casing is in the cold position, when the inner casing is in the hot position, and during periods of transition.

By making the depth of the grooves 42 equal to the anticipated maximum vertical movement or deflection of the seal member, misalignment of the seal member in the hot position is obviated.

If desired, as illustrated in FIGS. 2 and 3, the seal member 40 may be further supported in additional grooves or recesses 60, of similar shape to grooves 42, provided by a pair of longitudinally extending retainer members 61 secured to one pair of the shroud rings 28.

In the appended claims, terms such as "above" or "below" are utilized for convenience only and are not intended as limitations.

While the invention has been shown in but one form, it will be obvious to those skilled in the art that it is not so limited, but is susceptible of various changes and modifications without departing from the spirit thereof.

What is claimed is:

1. In casing structure defining fluid pressure regions, said casing structure including an upper casing and a lower casing having juxtaposed terminal portions defining a longitudinally extending space, said terminal portions including continuous longitudinally extending surfaces adjacent said space, said terminal portions being movable toward each other in response to thermal expansion, a longitudinally extending deformable seal member straddling said space for restricting fluid flow through said space, said seal member having a longitudinally extending deformable sealing surface that is non-rectilinear in cross section and that projects in the lateral direction to one side of said continuous surfaces, said terminal portions including wall structure slidably retaining said seal member and biasing said sealing surface against said continuous surfaces and for restricting movement of said seal member in the direction opposite to said lateral direction, and said wall structure allowing movement of said seal member in the direction transverse to said lateral direction, whereby upon thermal expansion, as said terminal portions approach each other said seal member tends to straighten but remains biased by said wall structure against said continuous surfaces.

2. In casing structure defining fluid pressure regions, said casing structure including an upper casing and a lower casing having spaced terminal portions that define a longitudinally extending space, said terminal portions including continuous longitudinally extending surfaces disposed above and below said space, said terminal portions being movable toward and away from each other in response to thermal expansion and contraction of the casings, respectively, a longitudinally extending flexible seal member straddling said space for restricting fluid flow through said space, said seal member having a longitudinally extending flexible sealing surface portion that is of curved cross section and that projects laterally to one side of said continuous surfaces, said sealing member further having a pair of curved marginal portions extending from said sealing portion, said terminal portions including wall structure defining recesses, said seal member being slidably disposed in said recesses, said wall structure biasing said sealing surface against said continuous surfaces and restricting movement of said seal member in the direction opposite to said lateral direction, and said wall structure allowing movement in the direction substantially normal to said lateral direction, whereby upon thermal expansion as said terminal portions approach each other, said seal member tends to flex and straighten but remains biased by said wall structure against said continuous surfaces.

3. In tubular casing structure including an upper casing and a lower casing having spaced terminal portions that define a longitudinally extending joint, said terminal portions including continuous longitudinally extending surfaces disposed above and below said space, said terminal portions being movable toward and away from each other in response to thermal expansion and contraction, respectively, a longitudinally extending flexible and resilient seal member straddling said space for restricting fluid flow through said space, said seal member having a longitudinally extending flexible and resilient sealing surface that is substantially V-shape in cross section and that projects laterally to one side of said continuous surfaces with the apex of the V approximately centrally located between said continuous surfaces, wall structure defining a pair of opposed recesses having said seal member slidably received therein, said wall structure biasing said sealing surface against said continuous surfaces, regardless of expansion and contraction, said wall structure restricting movement in a direction transverse to the longitudinal axis and maintaining said seal member centrally positioned with respect to said space regardless of expansion of said terminal portions, and the furthest extremities of said recesses being spaced from each other a distance greater than the major cross-sectional dimension of said seal member to permit said seal member to flex and tend to flatten the V-shape as said terminal portions expand and approach each other and as said terminal portions contract and move away from each other to permit said seal member to flex and attain more curvature.

4. In an elastic fluid turbine, an outer tubular casing structure, an inner tubular casing structure defining a fluid flow path and jointly with said outer casing defining an annular chamber, said inner casing structure including an upper casing and a lower casing having juxtaposed terminal portions that define a longitudinally extending space, said terminal portions including continuous surfaces above and below said space, said upper casing and said lower casing being movable relative to each other in response to thermal expansion and contraction from a first spaced position to a second spaced position in which said ter-

minal portions are closer to each other relative to said first position, a longitudinally extending resilient seal member for restricting fluid flow between said flow path and said chamber through said space, said seal member having a longitudinally extending central sealing surface portion of curved cross section abutting said continuous surfaces, said seal member further having opposed marginal portions connected to said central portion and extending vertically therefrom, said upper and lower casings having grooves defined by wall structure, said seal member being partially disposed in said grooves and said marginal portions engaging said wall structure for biasing said seal member into sealing relation with said continuous surfaces, said wall structure and said continuous surfaces being slidable transverse to the longitudinal axis of said seal member, said wall structure including surfaces defining said grooves that are arcuate in cross section, tending to pivot said marginal portions of said seal member in opposite directions along longitudinal axes coinciding with said continuous surfaces, whereby said central surface portion tends to straighten upon movement from said first position to said second position while remaining biased against said continuous surfaces.

5. In an elastic fluid turbine, an outer tubular casing, an inner tubular casing divided into an upper casing and a lower casing and defining a longitudinally extending fluid flow path, said inner casing comprising a plurality of pairs of interconnected segmented rings, axially spaced and radially projecting wall members interconnected to said rings, said wall members connecting said inner casing to said outer casing, said rings being spaced from said outer casing and forming therewith a longitudinally extending annular chamber, means for supplying a pressurized fluid to said chamber; each of said pairs of rings having juxtaposed terminal portions forming aligned and continuous surfaces defining a longitudinal space; said terminal portions moving from a cold position to a hot position due to thermal expansion and contraction and modifying said space, said wall members having arcuate wall surfaces defining opposed and longitudinally extending open-ended grooves adjacent said space, and a longitudinally extending flexible seal member partially disposed in said grooves for restricting fluid flow between said flow path and said chamber, said seal member having a central sealing surface and spaced marginal portions together defining substantially a C-shape in cross section, said sealing surface being in contact with said aligned continuous surfaces at all times, said marginal portions being in contact with said arcuate walls for biasing said sealing surface against said continuous surfaces and in sealing relation therewith, said arcuate walls and said continuous surfaces being slidable relative to said seal member during movement of said casings incident to thermal expansion and contraction, said arcuate walls tending to pivot the marginal portions of said seal member in opposite directions along longitudinal axes coincident with said continuous surfaces, whereby the C-shape of said seal members tends to be straightened upon decrease in width of said space and tends to become more curved upon increase in width of said space.

6. In an elastic fluid turbine, a rotor supporting rotating blades, an outer casing, an inner casing divided into upper and lower casings defining a longitudinally extending fluid flow path, said inner casing comprising divided shroud rings for supporting stationary blades, divided bridging rings encompassing said rotating blades, and divided supporting rings to which are secured on opposite sides one of said shroud rings and one of said bridging rings; said shroud rings and said bridging rings being spaced from said outer casing and forming therewith a longitudinally extending annular chamber, means for supplying a pressurized fluid to said chamber; each of said rings including juxtaposed terminal portions forming a continuous longitudinal space, wall structure comprising inner and outer wall surfaces defining rows of longitudinally extending grooves having opposed access openings adjacent said space, said terminal portions also forming longitudinal continuous surfaces on opposite sides of said spaces, and an integral longitudinally extending resilient seal member disposed between said terminal portions; said seal member having a curved central sealing surface portion and curved marginal portions together forming a C-shape in cross section, said seal member having said marginal portions disposed in said grooves and in slidable abutment with said outer walls, and said central sealing portion being disposed in slidable abutment with said aligned continuous surfaces, thereby allowing movement of said terminal portions due to thermal expansion and contraction, said central portion extending to the side of said continuous surfaces opposite the side on which said outer walls are disposed when cold, said outer walls tending to pivot said marginal portions in opposite directions along longitudinal axes coinciding with said continuous surfaces while maintaining said central sealing surface biased against said continuous surfaces, whereby the C-shape of said seal member tends to be elongated upon movement from said cold position to said hot position and tends to become constricted upon movement from said hot position to said cold position.

References Cited in the file of this patent

UNITED STATES PATENTS

2,815,645	Downs	Dec. 10, 1957
2,848,156	Oppenheimer	Aug. 19, 1958

FOREIGN PATENTS

510,505	Great Britain	Aug. 2, 1939

United States Patent [19]

Hallinger et al.

[11] **Patent Number: 4,623,298**

[45] **Date of Patent: Nov. 18, 1986**

[54] **TURBINE SHROUD SEALING DEVICE**

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[73] Assignee: **Société Nationale d'Etudes et de Construction de Moteurs d'Aviation**, Paris, France

[21] Appl. No.: **652,475**

[22] Filed: **Sep. 20, 1984**

[30] **Foreign Application Priority Data**

Sep. 21, 1983 [FR] France 83 14974

[51] **Int. Cl.⁴** **F01D 11/02**

[52] **U.S. Cl.** **415/139**; 415/174

[58] **Field of Search** .... 415/174, 138, 139, 134, 415/135, 136, 137, 172 A, 173 R

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,510,734	6/1950	Bodger	416/190
2,997,275	8/1961	Bean et al.	415/135
3,532,437	10/1970	Strub	415/136
3,867,061	2/1975	Moskowitz	415/174
3,986,789	10/1976	Pask	415/139
4,076,455	2/1978	Stargardter	416/191
4,272,290	6/1981	Lesgourgues	415/174
4,295,785	10/1981	Lardellier	415/174
4,346,904	8/1982	Watkins, Jr.	415/174

FOREIGN PATENT DOCUMENTS

1816066	8/1969	Fed. Rep. of Germany .
1331030	5/1963	France .
1519898	2/1968	France .
2514409	10/1981	France .
91210	1/1920	Switzerland .
2072760	10/1981	United Kingdom .

OTHER PUBLICATIONS

Seals for Rotating Shafts, Dr. E. S. Moult, Sep. 1975, pp. 727-731 *Engineering*.

Primary Examiner—Everette A. Powell, Jr.
Attorney, Agent, or Firm—Bacon & Thomas

[57] **ABSTRACT**

An improved guide vane shroud sealing device is disclosed wherein the shroud is formed from a plurality of segments, each segment having interengaging, "Z" shaped edges. The "Z" edges each have a mid-portion extending parallel to the rotational plane of a rotor blade wheel, and leading and trailing edge portions extending from this mid-portion to the leading and trailing edges of the shroud segments. A honeycomb packing structure seals the inner surface of the guide vane shroud in conjunction with labyrinth sealing fins on the rotor wheel. The honeycomb structure is oriented such that opposite sides of each cell which are joined to adjacent cells extend at an angle of approximately 60° to the rotational plane of the rotor blade wheel.

3 Claims, 6 Drawing Figures

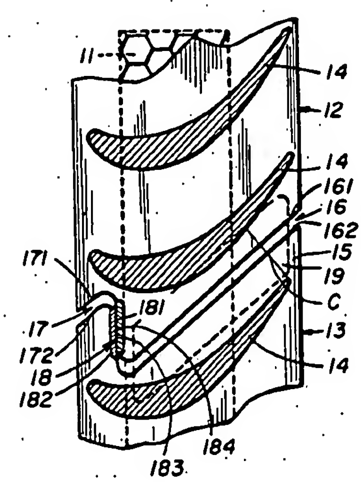


FIG. 1

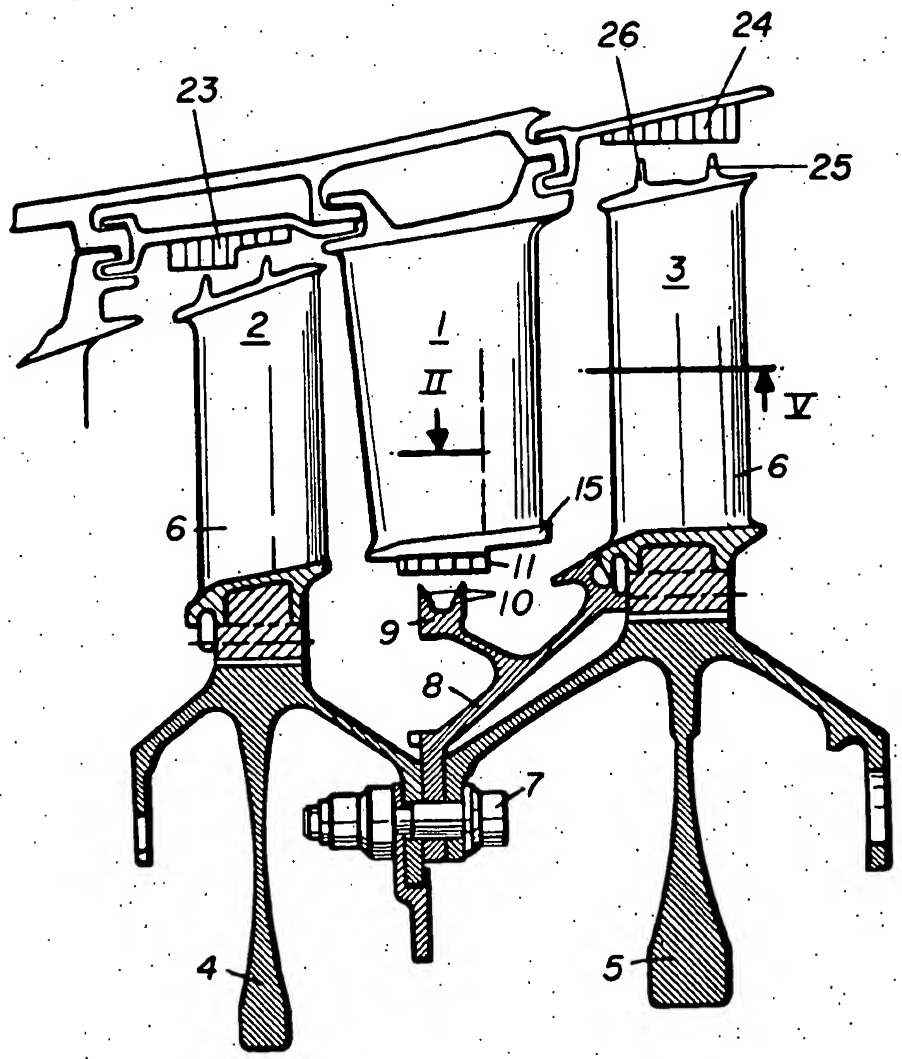


FIG.2

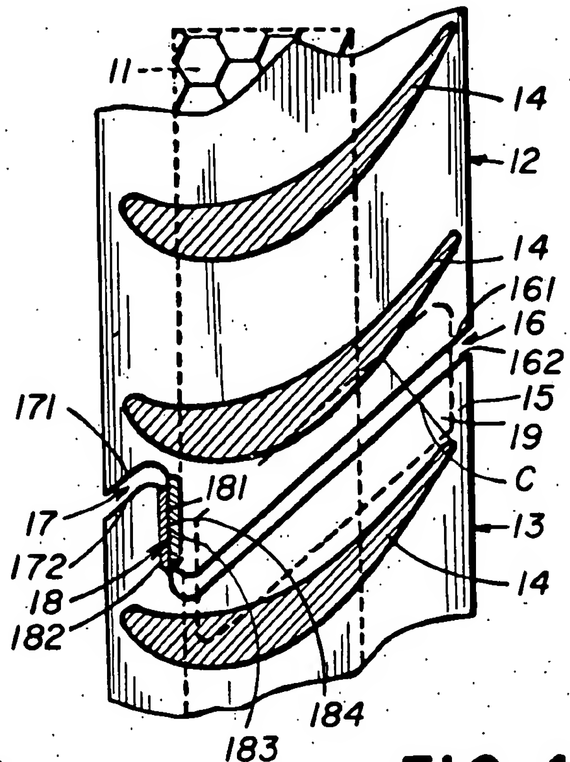


FIG.3
(PRIOR ART)

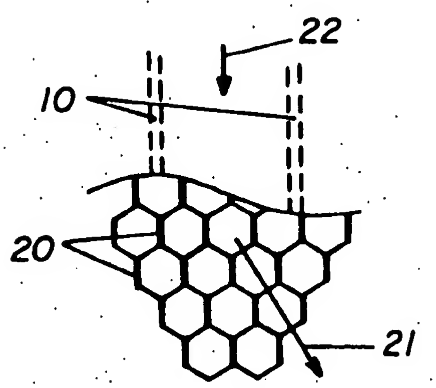


FIG.4

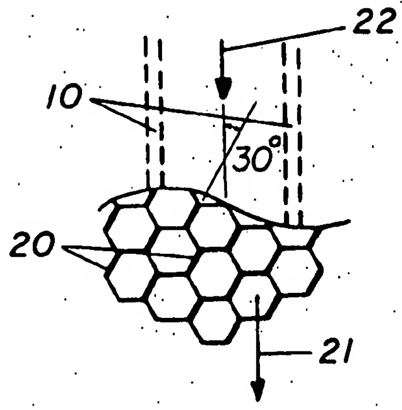


FIG. 5
(PRIOR ART)

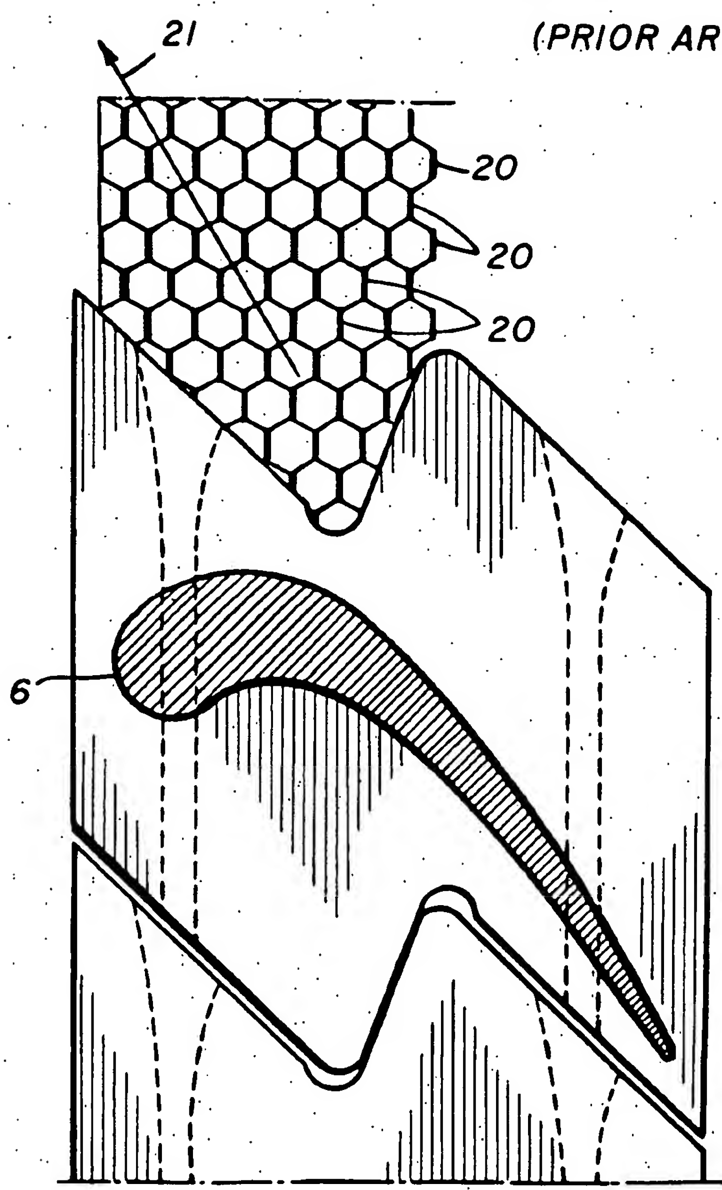
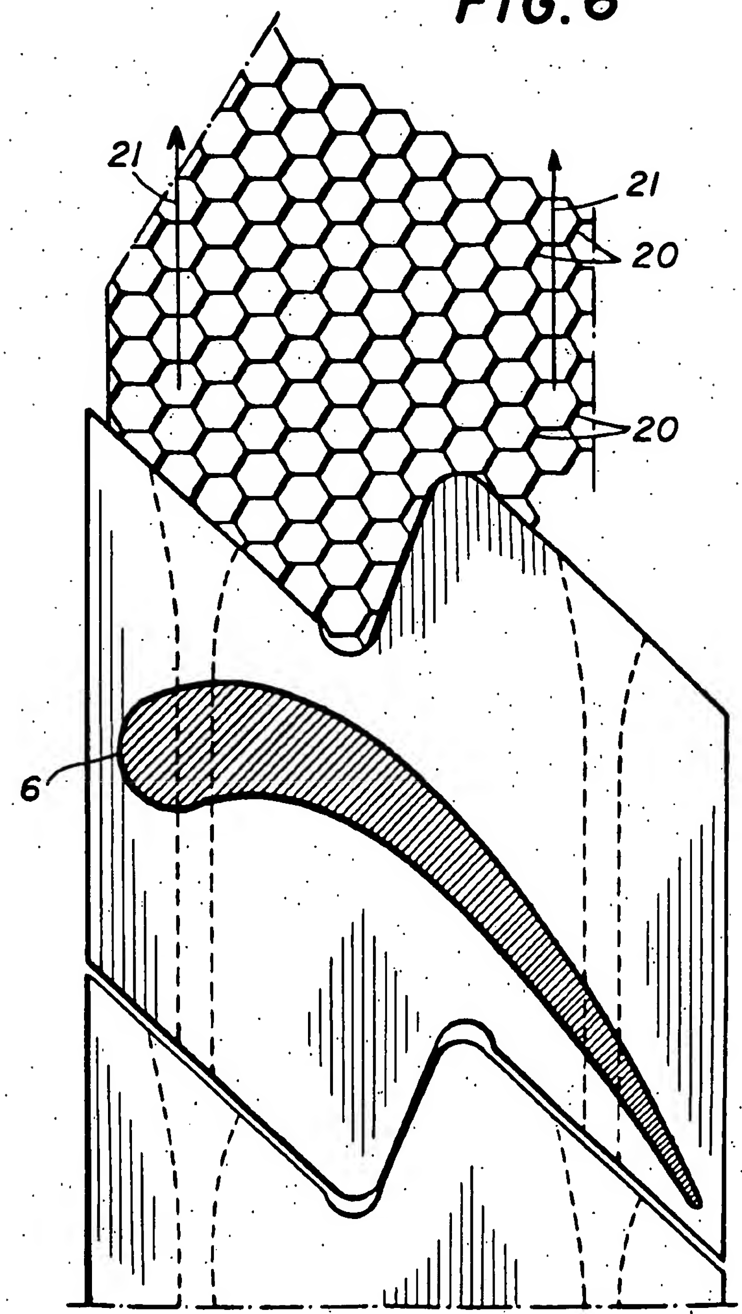


FIG. 6



TURBINE SHROUD SEALING DEVICE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The instant invention relates to an improved device for effecting a seal between a stationary guide vane structure and a rotor blade wheel.

2. Brief Description of the Prior Art

French Pat. No. 1,331,030 discloses a compressor vane ring wherein the individual vanes are fastened to inner and outer rings. The outer ring is formed by a plurality of arcuate segments having overlapping ends which are welded together. The overlapping portions serve as reinforcing ribs to increase the strength of the guide vane structure.

French Pat. No. 1,519,898 describes a rotor blade system wherein an outer ring connecting the blades is formed from a plurality of segments, each segment having generally "Z" shaped end portions. The end portions of one segment contact end portions of an adjacent segment at a mid-portion, while the portions on either side define a gap therebetween. The untwisting of the blades during their operation is utilized to provide a continuous connection between the blade segments on the rotor wheel.

French Pat. No. 2,514,409 discloses a rotor blade system wherein the blade wheel is formed from a plurality of segments. The inner, base portion of the segment is attached to a rotor disc, while the outer portion of the segment, which interconnects the tips of the blades, are interconnected by sealing means to prevent fluid leakage past the joint between adjacent segments. The sealing is affected by plate members inserted in correspondingly aligned slots in the ends of each segment.

The main object of the prior art in using the "Z" shape at the ends of the segments is to achieve a rigid and continuous connection between adjacent blades or blade segments when these are subjected to centrifugal action. Little, if any, consideration has been given to reducing the leakage of the gas passing across the turbine blades in a radial direction at the juncture of these segments, when the "Z" configuration is utilized.

SUMMARY OF THE INVENTION

The instant invention discloses a system for improving the seal between a guide vane shroud formed from a plurality of guide vane shroud segments and the rotating parts of the turbine assembly. This is accomplished by utilizing a specifically shaped "Z" geometry on each of the ends of the guide vane shroud segments which allows thermal expansion of the respected segments and at the same time avoids deleterious vibrations. Sealing means are provided between adjacent guide vane shroud segments to minimize radial leakage. A honeycomb-type labyrinth type seal is provided on the radially innermost sides of the guide vane shroud adjacent a labyrinth sealing fin extending from the rotor wheel structure. The honeycomb labyrinth seal is formed such that each of the hexagonal cells has two opposite sides joined to adjacent cells and is oriented in the turbine structure such that these opposite sides lie in parallel planes extending at an angle of approximately 30° to the rotational plane of the rotor blade wheel.

The "Z" shaped ends of each of the guide vane segments comprises a mid-portion, which extend generally parallel to the rotational plane of the rotor blade wheel, a leading edge portion which extends from the mid-portion to a leading edge of the segment, and a trailing edge portion which extends from the mid-portion to a trailing edge of the segment. Sealing plates may be installed in aligned slots such that it extends between trailing edge portions of each segment to minimize radial leakage.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a partial, side sectional view of a turbine utilizing the shroud sealing device according to the invention;

FIG. 2 is a partial sectional view in the direction of II in FIG. 1.

FIG. 3 is a partial view, also taken in the direction of arrow II in FIG. 1 showing the orientation of the honeycomb seal according to the prior art.

FIG. 4 is a partial view viewed in the direction of arrow II in FIG. 1 similar to FIG. 3, but showing the orientation of the honeycomb seal according to the present invention.

FIG. 5 is a partial cross-sectional view taken in the direction of arrow V in FIG. 1 showing the orientation of a honeycomb seal adjacent the ends of rotor blades according to the prior art.

FIG. 6 is a partial, sectional view taken in the direction of arrow V in FIG. 1 showing the orientation of a honeycomb seal adjacent a rotor blade wheel according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

FIG. 1 shows a partial, sectional view of a turbine having an outer housing, a fixed guide vane stage 1 and a pair of rotor blade stages 2 and 3 located on either side of the fixed guide vane stage. Each of the rotor blade stages comprise a plurality of turbine blades 6 attached to rotor wheels 4 and 5, respectively, by known means. These are retained within the housing so as to rotate about a common, rotational axis which extends generally parallel to the longitudinal axis of the housing. Rotor blade wheels 4 and 5 are attached to each other by bolts 7 extending through aligned flange structures associated with each wheel. A seal is provided between the radially innermost end portion of the guide vane stage 1 and the rotating turbine wheels 4 and 5 by labyrinth seal means 9 rigidly affixed to supporting flange 8. Labyrinth seal means 9 has a plurality of radially extending, labyrinth seal fins 10 extending therefrom which interact with honeycomb-type labyrinth seal packing 11, to be described in more detail hereinafter. Honeycomb packing structure 11 is affixed to inner ring 15 which interconnects the innermost ends of the guide vanes 14 in stage 1.

As best seen in FIG. 2, the guide vane stage 1 comprises a plurality of guide vane shroud segments each having a plurality of stationary guide vanes 14, an inner ring 15 and an outer ring (not shown). The outer ring is also attached to the turbine housing as shown in FIG. 1, to fixedly support the guide vane stage therein. FIG. 2 shows the intersection of adjacent guide vane segements 12 and 13, each formed with a generally "Z" shaped edge. As shown in this figure, each of the "Z" shaped edges comprises a mid portion 181 and 182, a leading edge portion 171 and 172 which extends from the respective mid portion to a leading edge of each of the segments, and a trailing edge portion 161 and 162, which extends from the mid-portion to a trailing edge of each segment. Mid-portions 181 and 182 extend gener-

ally parallel to the plane of rotation of the rotor wheels 4 and 5, respectively, and contact each other along line 18. This arrangement prevents the adjacent guide vane segments from moving axially with respect to each other, but does not prevent circumferential displacements due to thermal expansion and contraction.

Vibrations from both the air flow through the guide-vane stage and the rotation of the compressor stages may cause mid portions 181 and 182 to prematurely wear thereby affecting the sealing capacity between the guide vane segments. In order to minimize this possibility, mid portions 181 and 182 may be covered by a wear-resistant material, such as a cobalt-based alloy. This material may be in the form of small plates 183 and 184 welded to the respective contacting surfaces.

In their normal configuration, trailing edge portions 161 and 162 are separated by a gap 16. Similarly, leading edge portions 171 and 172 are separated by a gap 17. Gaps 16 and 17 allow the circumferential expansion and contraction due to the normal changes in operating temperatures encountered in the turbine. Leading edge portions 171 and 172 extend at an acute angle from the respective mid-portions, as do trailing edge portions 161 and 162. As noted in FIG. 2, the trailing edge portions 161 and 162 are substantially longer than the leading edge portions 171 and 172. The trailing edge portions may extend from an upstream edge of labyrinth seal structure 11 to the trailing edge of the guide vane segment. Leading edge portion 171 may also extend parallel to trailing edge portion 161. A similar relationship may exist between leading edge portion 172 and trailing edge portion 162. The substantially longer length of the trailing edge portions permits the use of sealing plate 19 between adjacent segments to minimize radial leakage between them. The adjacent edges of guide vane segments 12 and 13 define aligned grooves into which the sealing plate 19 extends. The sealing plate is slidably retained in the groove to allow thermal expansion and contraction of the adjacent segments 12 and 13. The only portion of the juncture of adjacent guide vane segments which is not positively sealed is gap 17 between the leading edge portions 171 and 172. Since this extends for only a very short distance and is upstream from the intake of the guide vane system, it does not cause substantial perturbations in the air flow.

The labyrinth seal means comprises a honeycomb packing structure 11 fastened to the innermost side of the inner ring 15 of the guide vane structure. As is well known in the art, the honeycomb packing structure may be formed by welding, brazing, or otherwise bonding a plurality of crimped metallic strips to form a honeycomb structure with a plurality of hexagonally shaped cells. The metal may be a stainless steel or other high-temperature alloy to withstand the operational temperatures of the turbine. As seen in FIGS. 3 and 4, each cell of the honeycomb packing structure 11 has two opposite sides 20 joined to adjacent cells so as to form the honeycomb structure. As noted specifically in FIGS. 3 and 5, in the prior art these opposite joined sides were oriented such that they extended generally parallel to the plane of rotation of the rotor wheels, as indicated by arrow 22. As a result of this orientation, a path of least resistance, denoted by arrow 21, extended in the direction of the walls which were of single thickness (i.e., not joined to adjacent walls) which subtended an angle of approximately 30° to the direction of rotation of arrow 22. Accordingly, the labyrinth sealing fins 10 when they contacted the honeycomb packing structure 11 would tend to follow the path 21 through the honeycomb and thereby create grooves in the honeycomb structure significantly wider than the width of the labyrinth sealing fins themselves. This resulted in eventual degradation of the sealing capacity of the labyrinth seal.

In order to avoid this drawback, the instant invention proposes to orient the honeycomb packing structure such that the joined walls lie in parallel planes extending approximately at an angle of 30° with respect to the plane of rotation of the turbine wheels (and, of necessity, the labyrinth sealing fins 10). As shown in FIGS. 4 and 6, with this orientation, the path of least resistance 21 extends parallel to the rotational path of the turbine wheel, denoted by arrow 22. This minimizes the size of the groove in the honeycomb structure formed by the labyrinth sealing fins 10 and allows them to penetrate the honeycomb sealing structure with less stress. Thus, the sealing affect achieved by the labyrinth seal is more effective over a period of time.

The same orientation of the honeycomb sealing structure may be utilized in external seals 23 and 24 to affect a seal between the tips of the rotor blades 6 and the engine housing. FIG. 5 shows the orientation of the prior art external seals wherein the joined walls of the honeycomb structure extend generally parallel to the plane of rotation. Again, labyrinth sealing fins 25 and 26 tend to follow the path of least resistance, denoted by arrow 21, thereby forming unnecessarily wide grooves in the honeycomb structure. By orienting the honeycomb structure such that the joined walls subtend an angle of approximately 30° with respect to the plane of rotation of the rotor blade wheels, as shown in FIG. 6, the path of least resistance, denoted by arrow 21, is parallel to the rotational plane of the rotor blade wheel.

The foregoing description is provided for illustrative purposes only and should not be construed as in anyway limiting this invention, the scope of which is defined solely by the appended claims.

What is claimed is:

1. In a turbine having an outer housing and at least one rotor blade wheel located within the outer housing having at least one labyrinth sealing fin extending radially therefrom, the improvements comprising:

(a) a plurality of guide vane shroud segments forming a guide vane shroud attached to the housing and located adjacent to the rotor blade wheel, each of the shroud segments having generally "Z" shaped ends on its radially inner ring, each such end having a mid-portion extending generally parallel to the rotational plane of the rotor blade wheel such that it contacts a corresponding mid-portion of an adjacent shroud segment, a leading edge portion extending at an acute angle from the mid-portion to a leading edge of the shroud segment and a trailing edge portion, longer than the leading edge portion, extending at an acute angle from the mid-portion to a trailing edge of the shroud segment, the leading and trailing edge portions of one segment being spaced apart from corresponding leading and trailing edge portions of adjacent shroud segments to allow for thermal expansion;

(b) a wear-resistant material attached to each mid-portion;

(c) seal means disposed between trailing edge portions of adjacent shroud segments; and,

(d) labyrinth seal means disposed on the radially innermost side of the guide vane shroud adjacent to the labyrinth sealing fin, the labyrinth seal means

comprising a honeycomb packing structure defining a plurality of hexagonal cells, each cell having two opposite sides joined to adjacent cells, such joined, opposite sides lying in parallel planes extending at an angle of approximately 30° to the rotational plane of the rotor blade wheel, the trailing edge portion extending approximately from an upstream edge of the labyrinth seal means to the trailing edge of the shroud segment.

2. The improved turbine of claim 1 wherein the wear-resistant material is a cobalt-based alloy.

3. The improved turbine of claim 1 further comprising second labyrinth seal means disposed on the turbine housing in the plane of rotation of the rotor blade wheel to effect a seal between the rotor blade tips and the housing, the second labyrinth seal means comprising a honeycomb packing structure defining a plurality of hexagonal cells, each cell having two opposite sides joined to adjacent cells, such joined opposite sides lying in parallel planes extending at an angle of approximately 30° to the rotational plane of the rotor blade wheel.

* * * * *

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 4,623,298

DATED : November 18, 1986

INVENTOR(S) : HALLINGER et al

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

Column 2, line 60, "segements" should be "segments".

Signed and Sealed this

Twenty-eighth Day of April, 1987

Attest:

DONALD J. QUIGG

Attesting Officer *Commissioner of Patents and Trademarks*

US005154581A

United States Patent [19]

Borufka et al.

[11] Patent Number: 5,154,581

[45] Date f Patent: Oct. 13, 1992

[54] **SHROUD BAND FOR A ROTOR WHEEL HAVING INTEGRAL ROTOR BLADES**

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[73] Assignee: Mtu Motoren- und Turbinen- Union Munchen GmbH, Munich, Fed. Rep. of Germany

[21] Appl. No.: 699,474

[22] Filed: May 13, 1991

[30] Foreign Application Priority Data

May 11, 1990 [DE] Fed. Rep. of Germany 4015206

[51] Int. Cl.[5] F01D 5/22

[52] U.S. Cl. 416/190; 416/191; 415/173.6

[58] Field of Search 415/173.1, 173.5, 173.6, 415/174.5, 139; 416/190, 191, 192

[56] References Cited

U.S. PATENT DOCUMENTS

2,945,673 7/1960 Hockert et al. 415/139
4,576,551 3/1986 Olivier et al. 415/173.6
4,623,298 11/1986 Hallinger et al. 415/173.6
4,710,102 12/1987 Ortolano 416/190
4,948,338 8/1990 Wickerson 416/190
4,986,737 1/1991 Erdmann 416/190

FOREIGN PATENT DOCUMENTS

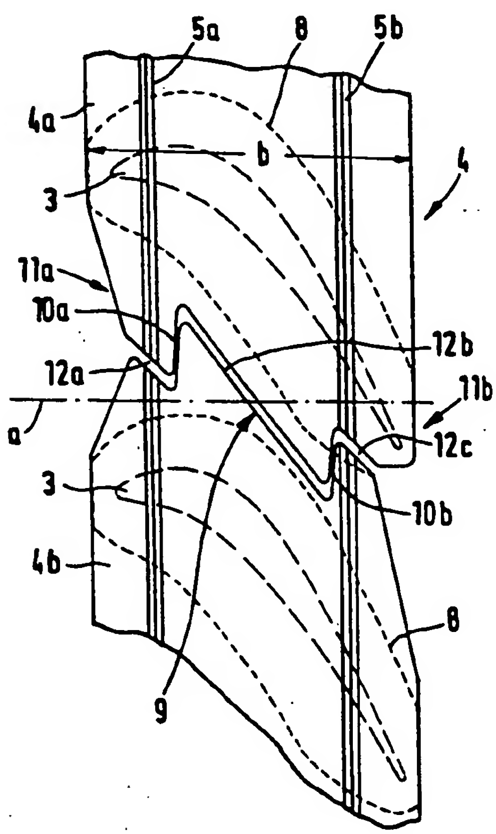
39807 3/1977 Japan 416/191

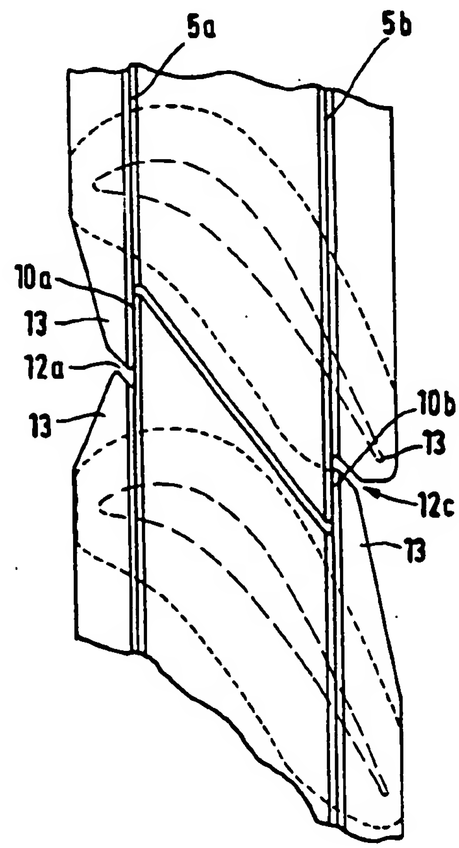
Primary Examiner—Thomas E. Denion
Assistant Examiner—Michael S. Lee
Attorney, Agent, or Firm—Ladas & Parry

[57] **ABSTRACT**

A shroud band for a rotor wheel having integral rotor blades wherein the shroud band has at least one Z-shaped separation gap at the band periphery, the Z-shaped gap having two parallel damping gaps which are axially spaced apart and which are inclined at an angle of from 70° to 90° relative to the axial direction. The damping gaps are narrow and are joined by an inclined wider free gap portion. During rotation, the damping gaps close up and the adjoining surfaces of the band come into frictional engagement whereas the free gap remains open.

17 Claims, 2 Drawing Sheets





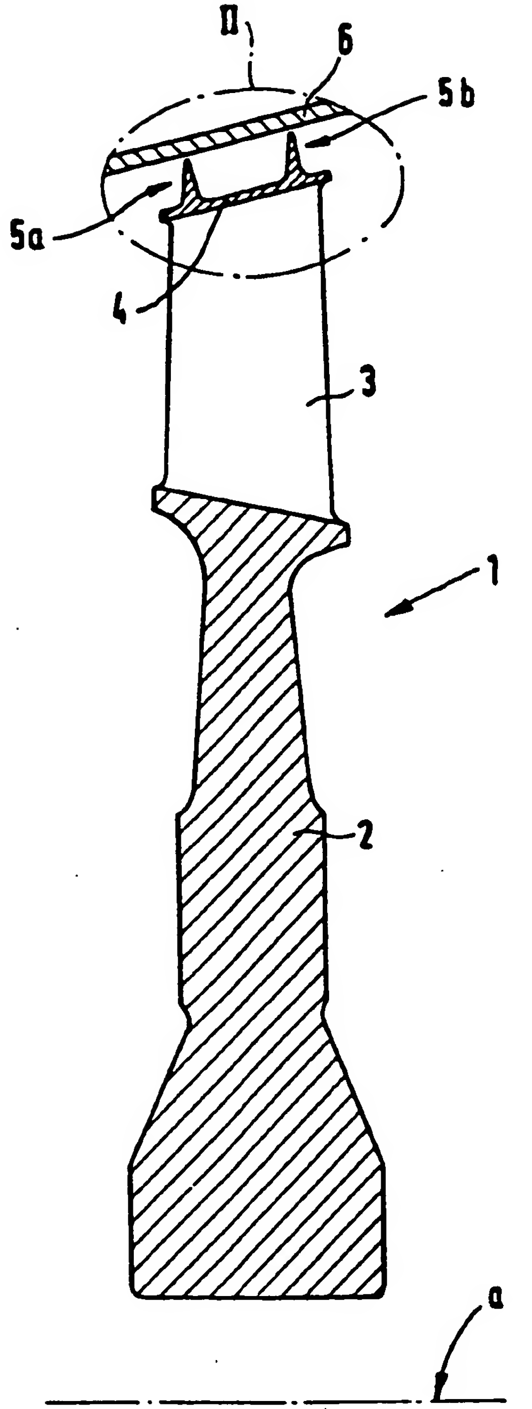
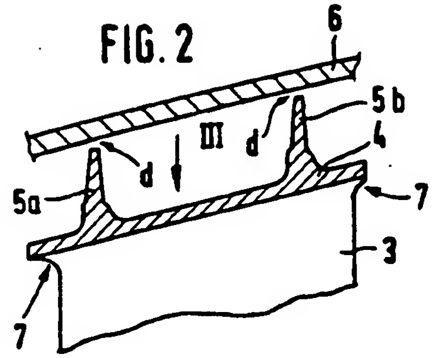


FIG. 1

FIG. 2



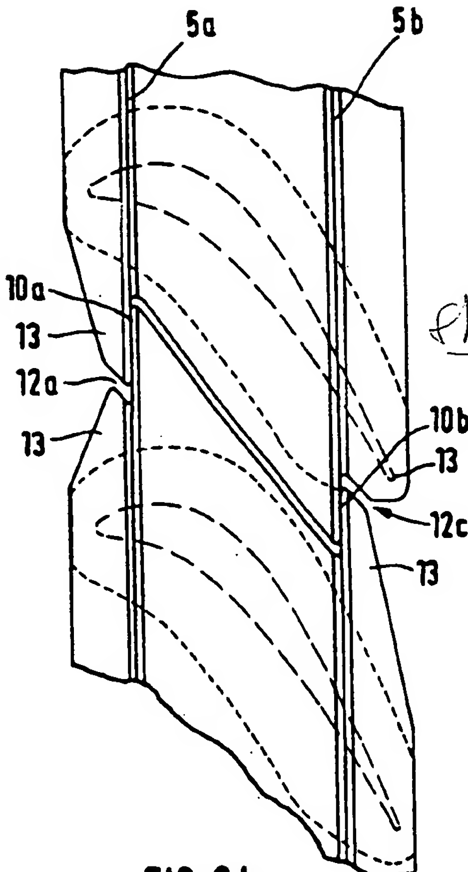


FIG. 3b

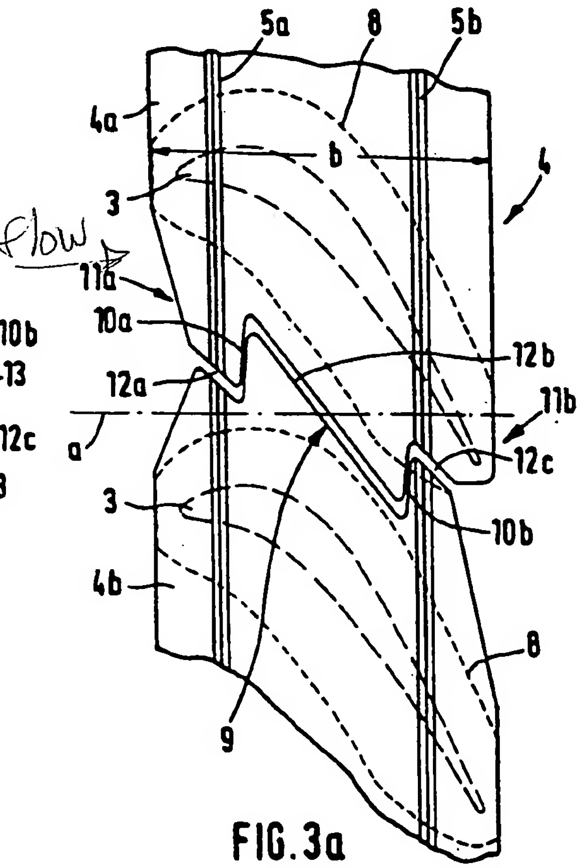


FIG. 3a

SHROUD BAND FOR A ROTOR WHEEL HAVING INTEGRAL ROTOR BLADES

FIELD OF THE INVENTION

The invention relates to a shroud band for a rotor wheel having integral rotor blades, the shroud band having at least one Z-shaped separation gap at the periphery of the blades, the Z-shaped separation gap having one part of relatively small gap width for damping purposes (damping gap) and another part of relative large gap width which remains open (free gap).

BACKGROUND AND PRIOR ART

In turbojet engines, the sealing gap between the rotating blades and the stationary engine housing represents a limiting variable which is of considerable importance for the efficiency of the engine. In order to minimize the sealing gap in turbines, it is known to provide the turbine with a shroud band which is attached to the blade tips. In the case of turbines with individually attached blades or blades attached in groups, the shroud band comprises band portions attached to the blades and having adjacent surfaces in toothed engagement with one another.

A shroud band constructed in this way is not possible with a turbine wheel having integral rotor blades. Conventional shroud bands for turbine wheels with integral rotor blades have a number of separation gaps for every three to five blades to compensate for expansions occurring during intermittent operation and to avoid development of additional stresses in the already highly stressed blades. These separation gaps extend substantially parallel to the chord lines of the adjacent blades in order to disconnect the shroud band from the blade tips in a way to minimize stress at the connection of the shroud band to the blade tips. In addition the separation gaps are formed with a short middle portion extending substantially at right angles to two end portions, so that the gaps have a substantially Z-shaped outline. The middle gap portion has a small width which is intended to be taken up, in operation, by thermal expansion, so that the adjacent shroud band portions are frictionally engaged at the middle portions of the gaps. In this way, vibration of the blades and of the shroud band portions connected thereto are effectively damped by friction, in which case the vibrations are additionally de-tuned. The two end or lateral portions of the gap (so-called free gaps) joined to the middle portion of the gap have sufficiently wide gap widths so that contact does not occur there under any conditions.

The conventional shroud band construction has the disadvantage that the individual middle gap portions cannot be produced with the small gap width desired to achieve optimum operating conditions, since the cutting tools used must have adequate rigidity and consequent thickness. For manufacturing reasons it is only possible to produce a finite gap which cannot be made less than from about 0.2 to 0.3 mm. This results in poorer damping and de-tuning characteristics of the vibrations which are developed. Consequently, vibrations of large amplitude occur, resulting in considerable stressing of the materials of the blades and shroud band.

SUMMARY OF THE INVENTION

An object of the invention is to provide an improved shroud band of the type described such that, on the one hand, an unrestricted peripheral or circumferential displacement of the shroud band portions is possible, so as to achieve compensation of thermal expansion with minimal stress and, on the other hand, the separation gap in the portions in contact during operation are made so narrow to provide adequate frictional damping of the vibrations. In addition, the shroud band is independent of different expansions of the rotor disc, i.e. is free on the periphery of the blade tips.

This object is attained according to the invention in that two damping gaps are provided which are axially spaced apart and which are oriented at an angle of from 70° to 90° relative to the axis of rotation of the rotor. The angle preferably is from 80° to 85°.

The essential advantages of the invention are that, while retaining an unchanged size of contact area between adjacent shroud band portions, a narrower gap is present and a stiffening of the arrangement is possible, since the frictional contact through the two parallel faces is possible. As a whole, the vibrations occurring in the several blades connected by a respective shroud band portion are effectively damped, since the damping gap is formed in such a way that pressure contact occurs during operation. This very narrow damping gap is advantageously independent of the circumferential expansion of the shroud band.

The increase in the angle of the two damping gaps with respect to the axis of rotation of the rotor to a value of between 70° and 90° has the effect that the circumferential expansion of the shroud band occurring, for example, during acceleration of the engine can take place without obstruction. In addition, peripheral or circumferential expansion affects the damping effect at the damping gap only slightly, i.e. the damping action is substantially independent of the operating condition of the turbine.

A further advantage is that the twisting of the blades occurring during operation due to centrifugal forces can be utilized to produce an effective reduction of the damping gap or the development of a damping thrust force, since according to the invention the angle of the damping gaps has such a large value.

In addition, it is advantageous that the very narrow damping gap can be made easier and more precise by doubling the number of damping gaps and thereby halving the length of each of the two damping gaps, which in turn leads to optimization of the shroud band portions by the shorter length of the damping gaps.

An advantageous further development of the invention provides that two continuous sealing lips spaced apart axially are integrally formed on the outer periphery of the band portions, and the damping gaps are arranged in the sealing lips. Radial sealing lips extending circumferentially around the shroud band are known, in order to define a remaining sealing gap between the shroud band and the outer housing of the flow duct, and fluid flow through the sealing gaps can be minimized by the spaced arrangement of the sealing lips one behind the other. In the event of a separation of the sealing lips, gaps or spaces are formed in the sealing lips, so that fluid can flow through the lips as well as through the separation gap between the lips and the housing. By virtue of the arrangement according to the invention in which the damping gaps extend in the sealing lips, these damping gaps are substantially reduced, or eliminated during operation. This advantageously results in a reduced fluid flow through the shroud band.

A further advantageous embodiment of the shroud band provides that the free gap portions in the region of the side edges of the shroud band are substantially widened in the manner of a wedge. In this way, overhang of the shroud band portions in the vicinity of the end regions of the free gaps, which are susceptible to vibration, can be reduced, so as to reduce the risk of vibration cracks of the shroud band.

BRIEF DESCRIPTION OF THE FIGURES OF THE DRAWING

The invention is further described with reference to two preferred embodiments illustrated in the attached drawing, in which:

FIG. 1 is a cross-section through a portion of a turbine having integral rotor blades;

FIG. 2 is an enlarged view of detail II in FIG. 1;

FIG. 3a is a plan view of one embodiment of a shroud band according to the invention as seen in the direction of arrow III in FIG. 2; and

FIG. 3b is a plan view of another embodiment of the shroud band according to the invention.

DETAILED DESCRIPTION

FIG. 1 shows in cross-section a turbine wheel 1 rotatable around an axis of rotation a, which essentially comprises a turbine disc 2 with integral blades 3 distributed around the periphery of the disc and a shroud band 4 integrally molded on the tips of the blades. The shroud band 4 comprises two axially spaced sealing lips 5a and 5b, which project radially outwards from a base portion of the band and define narrow sealing gaps with an outer casing 6 of the turbine. In FIG. 2 the sealing gaps are seen at d.

FIG. 3a shows a portion of the shroud band 4 as viewed in the direction of arrow III in FIG. 2. FIG. 3a shows two adjacent shroud band portions 4a and 4b, each of which is integrally molded on the tips of a plurality of respective blades 3, preferably three or four in number. In order that the stresses produced by the high centrifugal force in the transition regions 7 between the shroud band 4 and the blades 3 may be kept low, the blades are formed with regions 8 of large radii as shown by the boundary lines in FIG. 3a.

The adjoining surfaces of the edges of adjacent shroud band portions 4a and 4b define a separation gap 9 which essentially comprises two axially spaced damping gaps 10a and 10b and free gaps 12a, 12b and 12c. Free gap 12b extends between gaps 10a and 10b and gaps 12a and 12c extend from gaps 10a and 10b respectively towards the lateral side edges 11a and 11b of the shroud band. The free gaps 12a, 12b and 12c have widths of from about 0.6 to 0.8 mm, while the damping gaps 10a and 10b have widths of about 0.1 mm in the cold state.

During operation, the damping gaps 10a and 10b are normally completely closed, i.e. the shroud band portions 4a and 4b are frictionally engaged at these gaps whereas the free gaps 12a, 12b and 12c remain open.

The middle free gap 12b, which is situated between the two damping gaps 10a and 10b, is oriented substantially at an angle of 45° with respect to the axial direction a. The other free gaps 12a and 12c advantageously have substantially the same orientation. The damping gaps 10a and 10b, on the other hand, are oriented substantially at an angle of from 70° to 90° with respect to the axial direction a, and preferably at an angle of from about 80° to 85° to achieve the advantages according to the invention. The two damping gaps 10a and 10b are each located at a distance of substantially ¼ of the width b of the shroud band 4 from the side edges 11a and 11b of the shroud band. The free gap 12b extends between the damping gaps 10a, 10b in inclined relation thereto to form a Z-shape therewith.

The embodiment of the shroud band illustrated in FIG. 3b differs from the embodiment in FIG. 3a in that the two damping gaps 10a and 10b extend in the sealing lips 5a and 5b. This has the advantage that the gaps between adjacent portions of the sealing lips can be eliminated, during operation, by take-up of the gap width of the damping gaps 10a and 10b.

The end areas of the outer free gaps 12a and 12c at the side edges 11a and 11b are substantially enlarged in the manner of a wedge, so that overlying platform regions 13 of the band portions (which have a pronounced tendency to vibrate during operation) are reduced in size, without adversely affecting the operation of the shroud band 4. In this way, the mass of the unsupported platform regions is advantageously reduced.

Although the invention has been described in relation to specific embodiments thereof, it will become apparent to those skilled in the art that numerous modifications and variations can be made as disclosed in the attached claims.

What is claimed is:

1. A shroud band for a rotor wheel having a disc with a plurality of integral rotor blades extending circumferentially around this disc, the rotor wheel being rotatable about an axis of rotation, said shroud band comprising a plurality of band elements each secured to a respective plurality of rotor blades at tips thereof, adjacent band elements having adjoining facing edge surfaces forming Z-shaped separation gaps therebetween, each Z-shaped separation gap having two substantially parallel parts axially spaced from one another joined by an inclined part, said parallel parts of each Z-shaped gap extending at an angle of 70° to 90° with respect to said axis of rotation of the rotor wheel, the facing surfaces of the adjacent band elements at said parallel parts of said gap being relatively closely spaced as compared to the spacing of said surfaces in said inclined part of said gap.

2. A shroud band as claimed in claim 1 wherein said inclined part of said gap extends at an angle with respect to said axis of rotation of said wheel.

3. A shroud band as claimed in claim 2 wherein said band elements have side edges disposed circumferentially around said axis of rotation, said separation gap including inclined parts extending between said side edges an said parallel parts of said separation gap.

4. A shroud band as claimed in claim 2 wherein each band element includes a base portion affixed to the tips of the associated plurality of rotor blades, and to continuous sealing lips projecting radially from said base potion and extending circumferentially along the band element from one edge surface to the other.

5. A shroud band as claimed in claim 4 herein said parallel parts of the Z-shaped gaps extend in said sealing lips.

6. A shroud band as claimed in claim 1 herein said parallel parts of said Z-shaped gap have a width of less than 0.1 mm.

7. A shroud band as claimed in claim 6 herein said inclined part of the Z-shaped gap has a width of 0.6 to 0.8 mm.

8. A shroud band as claimed in claim 1 wherein said band elements have side edges disposed circumferen-

tially around said axis of rotation, adjoining facing edge surfaces of adjacent band elements defining further gaps extending from the parallel parts of the Z-shaped gap to said side edges.

9. A shroud band as claimed in claim 8 wherein said further gaps are parallel to said inclined part of said Z-shaped gap.

10. A shroud band as claimed in claim 8 wherein said further gaps extend at an angle of 45° with respect to said axis of rotation of the wheel.

11. A shroud band as claimed in claim 8 wherein said band elements form wedge-shaped openings extending in widening fashion from said further gaps to said side edges.

12. A shroud band as claimed in claim 1 wherein said parallel parts of each Z-shaped gap have centers disposed approximately at a distance of about ¼ of the width of the band from the side edges of said band.

13. A shroud band as claimed in claim 1 wherein said angle of the parallel parts of each Z-shaped gap relative to the axis of rotation is between 80° and 85°.

14. A shroud band as claimed in claim 1 wherein said inclined part of said gap extends at an angle of about 45° relative to said axis of rotation.

15. A shroud band as claimed in claim 1 wherein said parallel parts of the Z-shaped gap are dimensioned to be closed during rotation of the rotor wheel and produce frictional effects whereas said inclined part of the Z-shaped gap is dimensioned to remain open during rotation o the rotor wheel.

16. A shroud band as claimed in claim 1, wherein said shroud band has side edges and said parallel parts of the separation gap are axially spaced from said side edges.

17. A shroud band as claimed in claim 16, wherein said parallel parts of each separation gap are substantially equally spaced from said side edges.

* * * * *

United States Patent [19]

Jorgensen

[11] **Patent Number: 4,878,811**

[45] **Date of Patent: Nov. 7, 1989**

[54] **AXIAL COMPRESSOR BLADE ASSEMBLY**

[75] Inventor: **Stephen W. Jorgensen**, West Palm Beach, Fla.

[73] Assignee: **United Technologies Corporation**, Hartford, Conn.

[21] Appl. No.: **270,994**

[22] Filed: **Nov. 14, 1988**

[51] **Int. Cl.**[4] **F01D 5/22**

[52] **U.S. Cl.** .... 416/215; 416/190; 416/193 A

[58] **Field of Search** .... 416/215–218, 416/193 A, 190, 191

[56] **References Cited**

U.S. PATENT DOCUMENTS

787,907	4/1905	Farwell	416/215 X
2,148,653	2/1939	Semar .	
2,391,623	12/1945	Heppner	416/191
2,805,838	9/1957	Pickup	416/216 X
2,955,799	10/1960	Oickle	416/190 X
3,014,695	12/1961	Rankin et al.	416/193 A X
3,810,711	5/1974	Emmerson et al.	416/97
3,923,420	12/1975	Chifos	416/190
4,165,949	8/1979	Riollet	415/77
4,280,795	7/1981	Trousdell	416/218
4,465,432	8/1984	Mandet et al.	416/215
4,688,992	8/1987	Kirkpatrick et al.	416/215
4,767,273	8/1988	Partington	416/191

FOREIGN PATENT DOCUMENTS

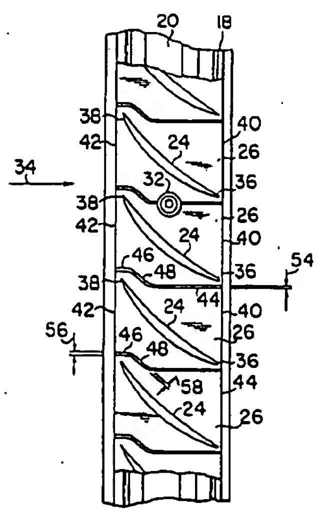
564048	9/1958	Canada	416/215
436567	11/1926	Fed. Rep. of Germany	416/215
1085643	7/1960	Fed. Rep. of Germany	416/215
985327	12/1982	U.S.S.R.	416/191
137379	5/1923	United Kingdom	416/215
2156908	10/1985	United Kingdom	416/215

Primary Examiner—Everette A. Powell, Jr.
Attorney, Agent, or Firm—Edward L. Kochey, Jr.

[57] **ABSTRACT**

Blade platforms (26) for compressor blades with airfoils (22) in high solidity relationship have edges of a major portion (44) a minor portion (46), and a canted intermediate portion (48). Minimum clearance (54) exists between the major portions (44) whereby twisting during stackup or operation is avoided. A circumferential seal (60, 64) is located under the platform coincident with major portion (44).

7 Claims, 3 Drawing Sheets



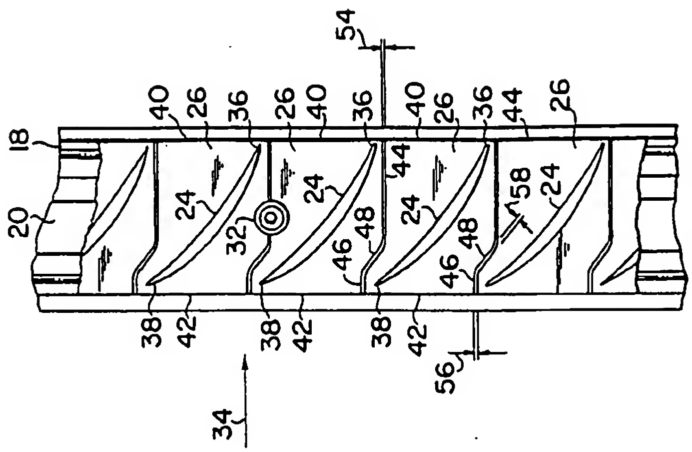


FIG. 5

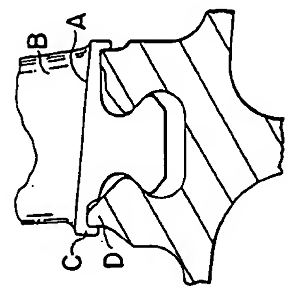


FIG. 3
PRIOR ART

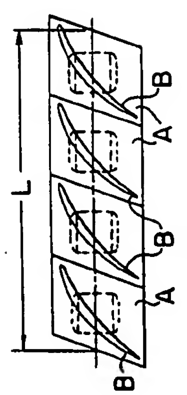


FIG. 1
PRIOR ART

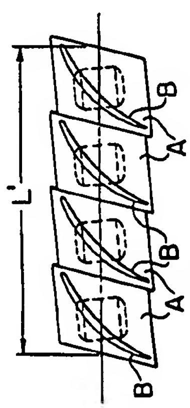


FIG. 2
PRIOR ART

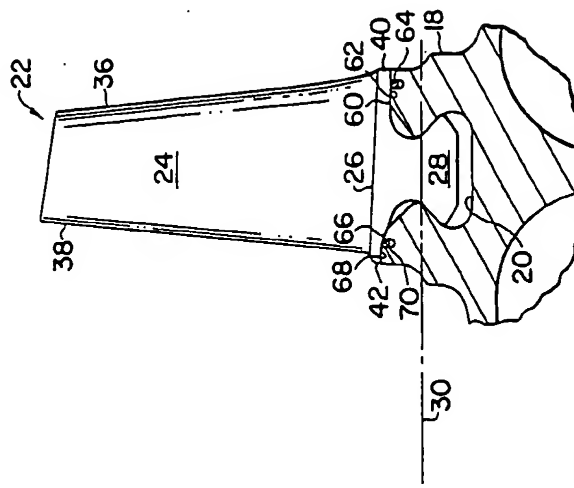


FIG. 6

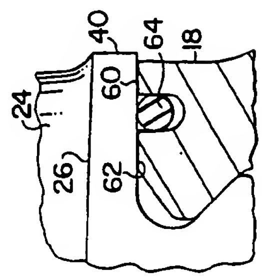
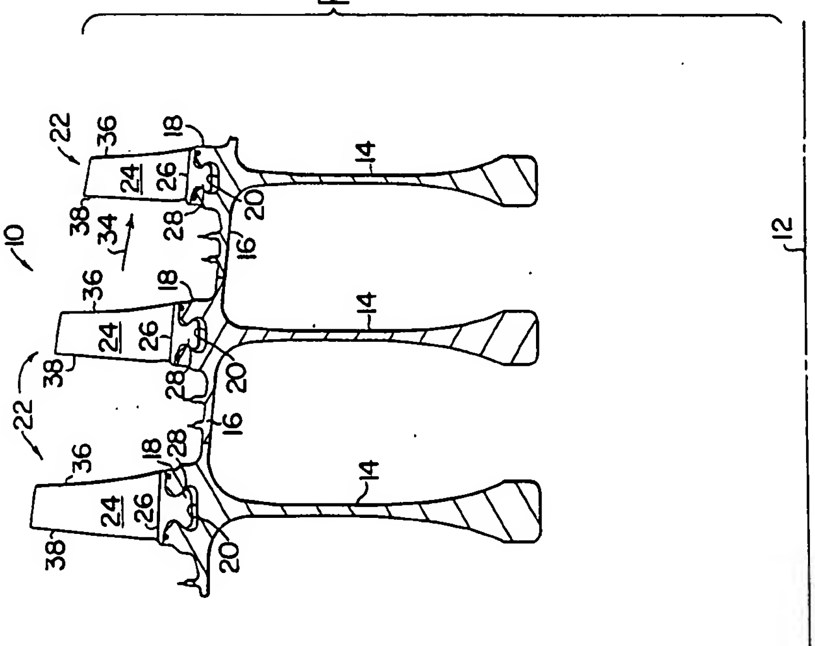


FIG. 7

FIG. 4



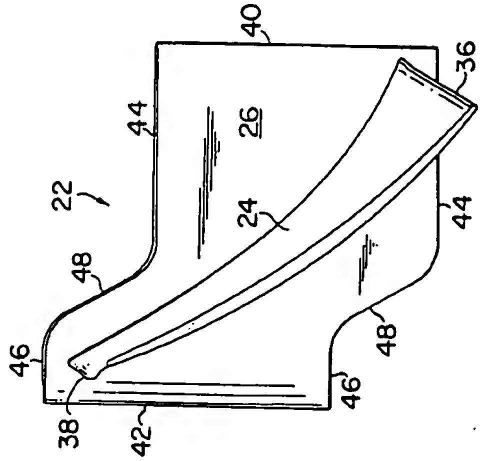


FIG. 8

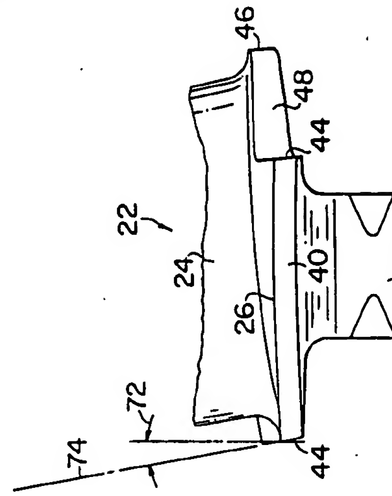


FIG. 10

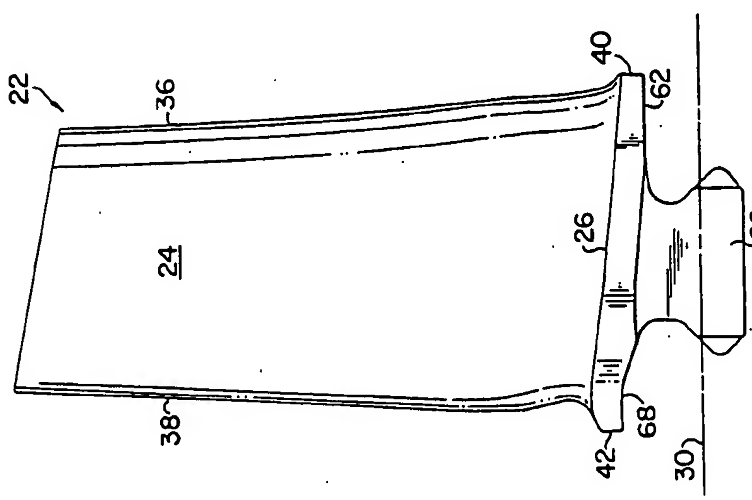


FIG. 9

AXIAL COMPRESSOR BLADE ASSEMBLY

The Government has rights in this invention pursuant to a contract awarded by the Department of the Air Force.

DESCRIPTION

1. Technical Field

The invention relates to axial compressors and in particular to blade assemblies therefore.

2. Background

In many cases compressor blades of a multi-stage axial compressor are secured to the rotor disks with fir tree blade roots. The roots slide into axially extending dovetail slots. This construction is feasible when there is convenient axis to the slot area for machining.

Lighter rotors may be built using drum type construction. This, however, interferes with access to the slot area. Therefore, an alternate construction uses a circumferential slot in the rotor disk to hold the blade roots. The blades are each passed into the slot through a entry slot and slide around the circumference until a full array of blades is installed.

With rectangular blade platforms such construction may be satisfactorily effected. However, on occasions the compressor design dictates high solidity requirement of the compressor blades. This means that when looking in the axial direction the blades overlap. Accordingly, the blades do not fit on a rectangular platform and a skewed platform must be used such as illustrated by platform A supporting blades B in FIG. 1.

During assembly of the blades the platforms may twist as shown in FIG. 2. When this happens the circumferential dimension L reduces to L' resulting in looseness of the platforms. Accordingly, this results in indeterminate spacing of the blades, sometimes to such an extent that an extra blade may even be installed. Also, during operation the blades are subject to such twisting or looseness.

One attempt to cure this problem is shown in FIG. 3 where rails C engage rim D. These rails must be tight with very little clearance to resist the twist and also to seal against recirculating air leakage between the blade platform and the rim. On the other hand, they must have ample clearance to permit them to slide around the circumference for assembly. Such design is expensive to manufacture to the tight required tolerances.

SUMMARY OF THE INVENTION

In a bladed drum type axial compressor assembly a plurality of disks each have a rim with a circumferential blade root retention slot. A plurality of blades are installed in each slot with each blade having an airfoil, a blade platform, and a root. The roots are retained within the slots with the airfoils in high solidity relationship with each other. Each blade platform has two circumferentially oriented ends with a first major axial edge portion at the first end which is substantially perpendicular to the end. The second, minor axial edge portion is at the second end and substantially perpendicular to that end. A canted intermediate edge portion joins the first and second axial portions. The plurality of blades are assembled with a minimum clearance between the major axial portions as compared to the other portions.

With the minimum clearance being at the major axial portions, the platforms operate as do rectangular portions to resist twisting during stackup at assembly. The greater clearances at the other portions establish the major portion as the determinant surface without requiring high tolerance manufacturing of the platform edges. The major portion also operates to resist twisting during operation. With the edges of the blade platform being substantially perpendicular to the ends there is no acute angle point of the blade platform which would be subject to deleterious vibration.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 shows the prior art skewed blade platforms;

FIG. 2 shows the prior art blade platforms as they twist;

FIG. 3 shows the prior art blades with a guide rail;

FIG. 4 shows a compressor rotor of drum type construction;

FIG. 5 is a plan view of the airfoil and platform as installed in the disk;

FIG. 6 is a side elevation of an installed blade;

FIG. 7 is a detail of the under platform seal;

FIG. 8 is a plan view of a blade;

FIG. 9 is a side elevation of FIG. 8; and

FIG. 10 is a front elevation of FIG. 8.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIG. 4, a drum type compressor rotor 10 rotating around center line 12 is formed of a plurality of disks 14. These disks are joined by extensions 16 forming the drum type rotor.

Each disk has a rim 18 including a circumferential slot 20. A plurality of blades 22 are installed in each slot.

Each blade 22 includes an airfoil 24, a blade platform 26 and a root 28. The root 28 in conjunction with slot 20 is designed so that the root and the rim are at a set radial location by intersecting that Z plane 30.

The circumferentially extending slot 20 has at one location in its circumference radially oriented loading slots which permits the blade to be installed in the radial direction whereupon it is then slide around the circumference inside slot 20. Once all the blades are installed, one or more locks 32 are secured to prevent further movement of the series of blades. When the last blade is installed, it along with the already installed blades is moved over one-half a blade spacing whereby all blades are away from the loading slot.

Airflow is in the direction shown by arrow 34 with leading edge 38 overlapping trailing edge 36 as viewed in the axial direction, thereby forming the high solidity relationship discussed above. Each blade platform 26 has a circumferentially extending first end 42 at the leading edge of the platform and a circumferentially extending second end 40 at the trailing side of the platform. A first major axial edge portion 44 extends from end 40 substantially perpendicular to end 40 and preferably greater than half the axial extent of the platform. If this edge should deviate from perpendicular by a significant amount, one of the two angles of the blade platform would form an acute angle which is then subject to vibration. Accordingly, it is preferable to maintain this edge in the near perpendicular position. A second minor axial portion edge 46 is located at the leading edge of the platform and substantially perpendicular to end 42. An intermediate canted portion 48 joins the two axially extending portions.

Each platform is fabricated such that clearance 54 between edges 44 is always less than clearance 56 be-

tween edges 46 and clearance 58 between edges 48. This insures that on stacking contact will be formed by close clearance 54 with some opening remaining at the other portions. Accordingly, these other portions will not interfere with accurate precise stackup of the blade assemblies.

Air pressure is increased in passing through the compressor blades 22. Accordingly, it is possible for leakage to occur beneath the blade platforms resulting in recirculation of the air being compressed and accordingly a reduction in efficiency. It is desirable to avoid or minimize such recirculation. The rim 18 has a first circumferential seal surface 60 adjacent to slot 20 on a trailing edge side of the slot. Each blade platform has a first circumferentially extending seal surface 62 on the underside of platform 26 and coincident with the first major axial edge portion 44. A circumferential seal in the form of seal ring 64 is located to sealingly abut the seal surfaces on both the rim and the blade platforms.

It is on this trailing edge side of the platforms that there is minimal clearance between the platforms. The seal and seal ring located at this position accomplishes the maximum sealing because of the minimum clearance between platform edges and accordingly minimum leakage between the platforms.

Rim 18 also has a second circumferential seal surface 66 adjacent to slot 20 on the leading edge side. Each blade platform also has a second circumferentially extending seal surface 68 on the underside of each platform and circumferential seal ring 70 is located between the two seal surfaces. This seal arrangement is located coincident with the second minor axial edge portion.

It can be seen in FIG. 10 that edge surfaces 44 while substantially extending in a radial direction are located with an angle 72 away from the precise radial direction. Edge surface 44 along with edge surfaces 46 and 48 are preferably formed by grinding in a single pass. Because of the potential extension of airfoil 24 beyond the edge of the platform, use of a precisely perpendicular edge would create interference between the grinding wheel and the blade. Accordingly, the edge portion 44 is formed off of the precise radial direction in an amount such that extension 74 of this surface clears all portions of airfoil 24.

In assembling the bladed rotor disk the root of each blade is passed through a radial entry slot and passed circumferentially around the disk with the root engaging the circumferential slot at the Z plane. This is continued until all but the final blade is installed. At this point the remaining gap is measured and compared with the width of the remaining blade. An appropriate final blade is selected with the blade platform producing a final gap between 0 and 0.02 inches. All blades are then slid around an additional half spacing and locked in place.

Since the blade platforms interact at the major axial edge portion, they do not twist during stackup and accordingly precise tolerances can be maintained. The same substantially axial edge portion interacts with the adjacent edge portions during operation to minimize twisting at that time. The use of the perpendicular intersection at the ends of the platform avoid acute angles producing fingers subject to vibration.

I claim:

1. A bladed drum type compressor assembly comprising:

a plurality of disks each having a rim with a circumferential blade root retention slot therein;

a plurality of blades installed in each slot, each blade having an airfoil, a blade platform supporting said airfoil, and a root supporting said blade platform;

said blade roots retained within said slots with said airfoils in high solidity relationship with each other;

each blade platform having two circumferentially oriented ends, a first major axial edge portion adjacent a first end and substantially perpendicular to said first end,

a second minor axial edge portion adjacent to the second end and substantially perpendicular to said second end, a canted intermediate edge portion intermediate said first and second axial edge portions; and

said plurality of blades assembled with the minimum clearance between adjacent platforms occurring between said first major portions as compared to said second minor portions and said canted intermediate portions.

2. An apparatus as in claim 1:

said major axial portions being greater than one-half he axial extent of said platforms.

3. An apparatus as in claim 1:

said first major axial portions being located at the trailing edge of said blade platform.

4. An apparatus as in claim 3:

said rim having a first circumferential seal surface adjacent said slot on the trailing edge side of said slot;

said blade platforms each having a first circumferentially extending seal surface on the underside of said platforms at an axial position coincident with said first major axial portion; and

a circumferential seal between said first circumferential seal surface of said rim and said first circumferentially extending surface of each blade platform.

5. An apparatus as in claim 4:

said rim also having a second circumferential seal surface adjacent said slot on the leading edge side thereof;

said blade platforms each also having a second circumferentially extending seal surface on the underside of said platforms at an axial position coincident with said second minor axial portion; and

a circumferential seal between said second circumferential seal surface of said rim and said second circumferentially extending surface of each blade platform.

6. An apparatus as in claim 1:

said axial edge portions and said intermediate edge portion of each platform having the edge surface thereof extending in a substantially radial direction at an angle away from the precise radial direction in an amount such that an extension of said edge surfaces clears said airfoil.

7. An apparatus as in claim 4:

said axial edge portions and said intermediate edge portion of each platform having the edge surface thereof extending in a substantially radial direction at an angle away from the precise radial direction in an amount such that an extension of said edge surfaces clears said airfoil.

* * * * *

United States Patent [19]

Riedmiller et al.

[11] **4,177,004**

[45] **Dec. 4, 1979**

[54] **COMBINED TURBINE SHROUD AND VANE SUPPORT STRUCTURE**

[75] Inventors: **Charles F. Riedmiller**, Greenhills; **Melvin Bobo**, Cincinnati, both of Ohio

[73] Assignee: **General Electric Company**, Cincinnati, Ohio

[21] Appl. No.: **846,958**

[22] Filed: **Oct. 31, 1977**

[51] **Int. Cl.**[2] **F02C 7/12**; F02C 7/20

[52] **U.S. Cl.** **415/116**; 415/136

[58] **Field of Search** 415/115, 116, 117, 135, 415/136, 137

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,859,934	11/1958	Halford et al.	415/137
3,628,880	12/1971	Smuland	415/115
3,742,705	7/1973	Sifford	415/117
3,966,354	6/1976	Patterson	415/116
3,986,720	10/1976	Knudsen et al.	415/136

Primary Examiner—Louis J. Casaregola

Attorney, Agent, or Firm—Dana F. Bigelow; Derek P. Lawrence

[57] **ABSTRACT**

A single turbomachine support structure provides both close tolerance radial positioning of a turbine shroud and close tolerance axial positioning of a vane. Radial movement of the vane without attendant radial movement of the shroud is accommodated by the use of surfaces designed to minimize constraint between the vane and the support structure. The linkage is further provided with cooling means to isolate thermal variations proximate the vane end thereof from the support structure end thereof.

8 Claims, 5 Drawing Figures

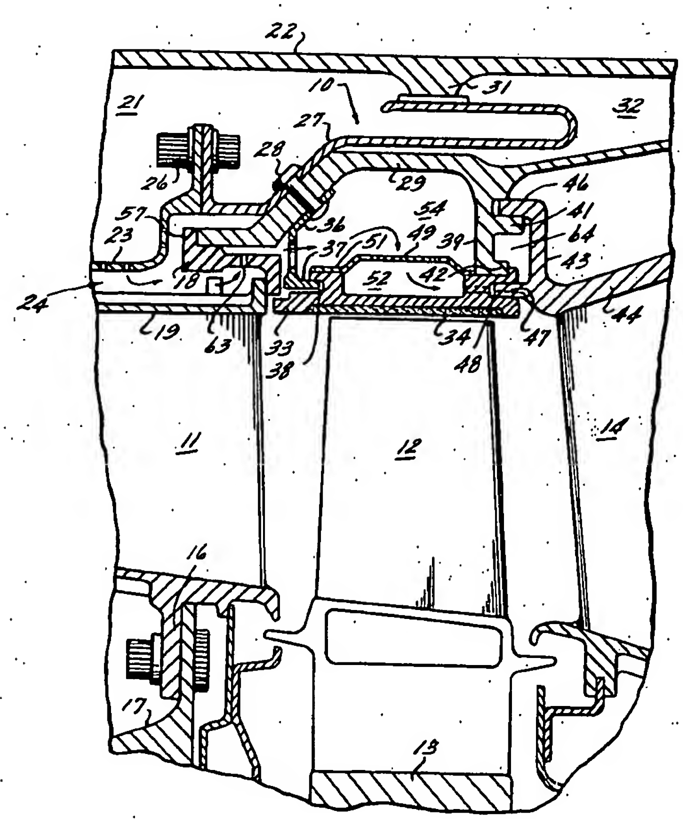


FIG 1

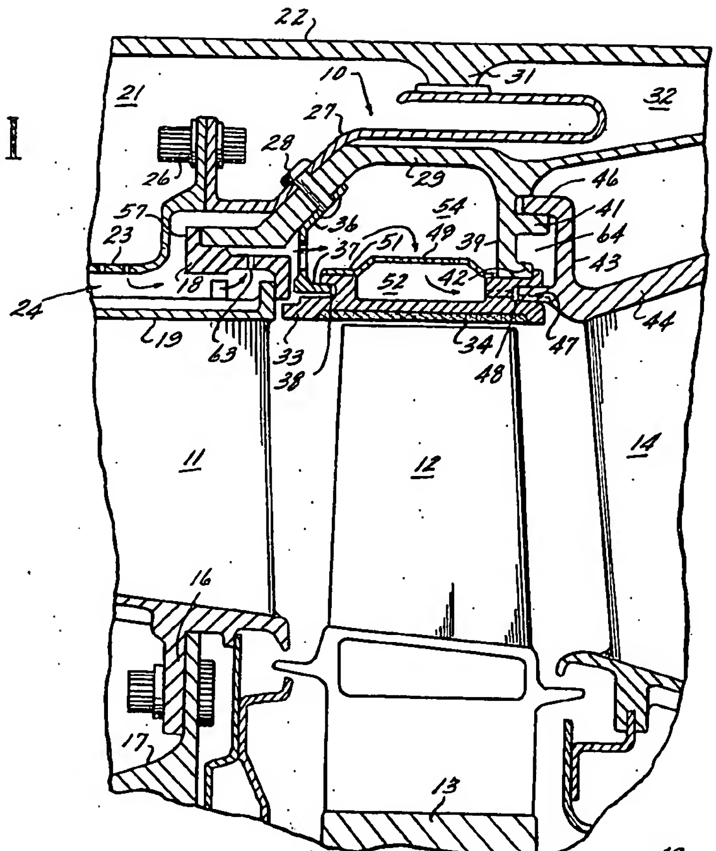


FIG 2

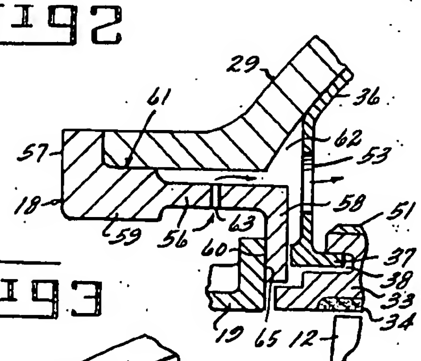


FIG 4

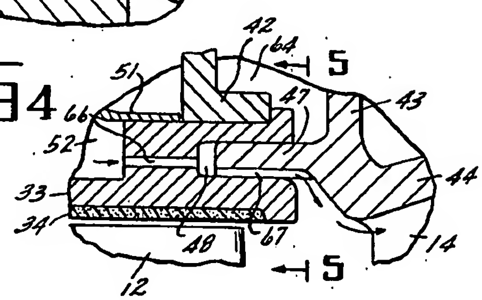


FIG 3

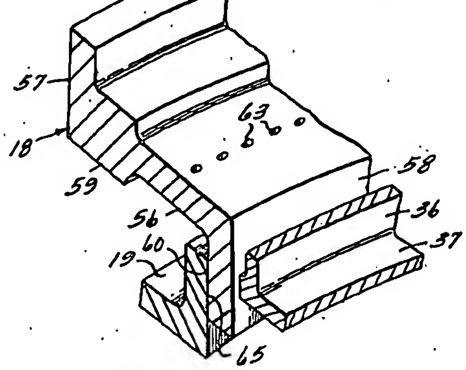
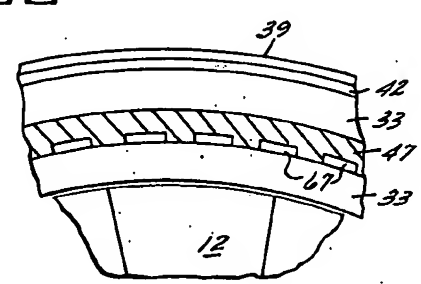


FIG 5



COMBINED TURBINE SHROUD AND VANE SUPPORT STRUCTURE

BACKGROUND OF THE INVENTION

This invention relates generally to gas turbine engines and, more particularly, to means for providing radial support for the turbine components of a gas turbine engine.

Turbine efficiency is largely determined by the ability to maintain a close clearance relationship between rotating blades and the surrounding stationary shrouds. Accordingly, in order to achieve and maintain good performance, it is necessary to have well maintained and repeatable close tolerances between those components. Wide clearances represent performance losses; and interference relationships, which may be caused by circumferentially nonuniform mechanical or thermal radial loads, tend to cause nonuniform wear of the shroud which subsequently may result in degraded performance. It is therefore desirable that the shroud be kept round and relatively isolated from such circumferentially nonuniform mechanical and thermal radial loads.

In an effort to limit the number of parts and simplify construction of a gas turbine engine, it is desirable to have a single support structure for supporting both the turbine shroud and the turbine vane immediately upstream thereof. For the reasons stated hereinabove, it is also desirable to have close tolerance radial positioning of the shroud. The vane, on the other hand, requires support only in the axial direction since the radial positioning thereof can be provided by other structure. However, because of the high axial loading, and especially at high speeds, any radial movement of the vane may be transmitted to the support structure and hence to the supported shroud to thereby effect its critical position. Nonuniformity in the circumferential shape of the shroud can thus result from nonuniform radial loads (as caused by either mechanical or thermal disturbances) which are transmitted from the vane to the shroud by way of the support structure.

It is therefore an object of the present invention to provide a turbine shroud support structure which maintains close radial tolerances and circumferential uniformity of the shroud.

Another object of the present invention is the provision for a support structure which provides both close tolerance radial positioning of a turbine shroud and axial positioning of an associated vane.

Yet another object of the present invention is the provision in an axial support structure for a turbine vane for preventing radial movements from being transmitted from the vane to the support structure.

Still another object of the present invention is the provision in an axial support structure for a vane for the isolation of certain portions of the support structure from thermal gradients which may occur at other portions thereof.

A further object of the present invention is the provision for a joint vane and shroud support structure which is economical to manufacture and effective in use.

These objects and other features and advantages become more readily apparent upon reference to the following description when taken in conjunction with the appended drawings.

SUMMARY OF THE INVENTION

Briefly, in accordance with one aspect of the invention, a nozzle support element is connected to the turbine shroud to provide close tolerance radial positioning therefor and is connected to the vane by a linkage which provides close tolerance axial positioning of the vane but permits radial movement thereof without attendant movement of the nozzle support element.

By another aspect of the invention, the flexible linkage comprises a ring which transmits both axial and radial loads toward the nozzle support element.

By yet another aspect of the invention, a plurality of circumferentially spaced holes are provided in the ring to provide fluid communication of cooling airflow through the ring to the outer surface of the shroud. In this way, the upstream end of the ring, which engages the nozzle support element is relatively isolated from the thermal variations which may exist at the downstream end of the ring adjacent the vane structure.

In the drawings as hereinafter described, a preferred embodiment is depicted; however, various other modifications and alternative constructions can be made thereto without departing from the true spirit and scope of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a fragmented axial sectional view of the turbine and vane portion of a turbomachine in which the present invention is embodied.

FIG. 2 is an enlarged view of the support ring portion thereof.

FIG. 3 is a fragmented perspective view of the support ring portion thereof.

FIG. 4 is an enlarged view of the C-clamp/shroud portion of the FIG. 1 apparatus.

FIG. 5 is a sectional view thereof as seen along line 5—5 of FIG. 4.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIG. 1, the invention is shown generally at **10** as incorporated into the turbine section of a gas turbine engine. A first stage high pressure nozzle or vane **11** receives high pressure gas from the combustor and directs it onto the first stage high pressure turbine blades **12** which act to convert the thermal energy into kinetic energy by way of rotating the turbine disk **13** in a conventional manner. The first stage nozzle **11** is bolted at its inner band **16** to the first stage support **17** which provides both radial and axial support therefor as well as forming the inner flow path wall from the compressor rear frame to the nozzle. Axial support is also provided at the outer diameter of the vane **11** by way of a ring **18** which engages the rear side of the outer band **19**. The hollow vanes **11** are cooled by compressor discharge air which enters the plenum **21**, which is defined on its outer side by the compressor rear frame **22**, and passes through an impingement plate **23** into a cavity **24** to enter the hollow blades and be discharged from a plurality of leading edge holes, gill holes and trailing edge slots in a manner well known in the art. The impingement plate **23** is secured by a plurality of bolts **26** to a seal **27** which in turn is secured by a plurality of bolts **28** to the stage 2 nozzle support **29**. The seal **27** is an annular device which extends radially outward to engage a pad **31** on the compressor rear frame **22** so as to thereby isolate from the plenum **21** a rear plenum

32 which contains cooling air at a lower pressure and temperature from that of the plenum 21.

Located radially outward from the row of turbine blades 12 is a plurality of circumferentially spaced shroud segments 33. The shroud segments 33 are closely spaced from the turbine blades 12 to prevent leakage of the hot gases therebetween, and contain a section of material 34 directly opposite the turbine blade row so constituted as to allow occasional interference between the components without attendant wear of the turbine blades with unstable wear conditions prevailing. Support for the shroud segments 33 is provided on the forward end by a plurality of shroud support segments 36 which are connected to the stage 2 nozzle support 29 by the fasteners 28, and which extend radially inwardly to terminate in an axial flange 37 which fits into a forward groove 38 of the shroud to provide positive radial placement of the shroud. Support at the rear end of the shroud segment is provided by a rim 39, forming an integral part of and extending inwardly from the stage 2 nozzle support 29 and having outer and inner flanges 41 and 42. Also forming a part of the support structure is a C-clamp 43 which forms an integral part of and extends forwardly of the outer band 44 of the second stage nozzle 14. The rim inner flange 42 abuts the outer side of the shroud segments 33 and the C-clamp 43 fits into the combination such that its outer flange 46 closely engages the outer side of the rim outer flange 41, and its inner flange 47 fits into the rear groove 48 of the shroud segment. In this way the C-clamp 43 holds together the shroud 33 and the stage 2 nozzle support 29 such that the shroud 33 moves radially with the nozzle support 29. Similarly, the forward end of the shroud 33 is also radially positioned with the nozzle support 29 by virtue of the rigid shroud support segment 36. Thus, the radial position of the shroud 33 is dependent on the radial position of the stage 2 nozzle support 29 which, in turn, is dependent primarily on its temperature. Generally, at lower speeds the compressor discharge air is cooler and the nozzle support 29 will assume a relatively inward position whereas at higher speeds, when the compressor discharge air is at a higher temperature, the nozzle support 29 will assume a relatively outward position. For steady-state operation, this accommodates a similar characteristic in the operational position of the turbine blades so as to result in the maintenance of a minimum clearance between the two components.

The shroud segments 33 are cooled by way of cooling air which enters a plurality of small holes 49 in an impingement plate 51 which is secured to the outer side of the shroud segment 33 by way of brazing or the like. The air then enters the cavity 52 and impinges on the outer side of the shroud segment 33 to cause cooling thereof. The cooling air originates in the compressor and passes through a plurality of ports 53 in the shroud support segment 36 to enter the plenum 54 surrounding the baffle plate 51.

Referring now to FIGS. 2 and 3, the ring 18 comprises an axially aligned section 56 and forward and rear radial sections 57 and 58, respectively, to define a generally S-shaped cross section. The axially aligned section 56 has an enlarged section 59 whose outer periphery forms a pad 61 for closely engaging with an interference fit, the nozzle support 29 at its inner side. At the same time, the rear surface of the radial section 57 fits tightly against the front surface of the nozzle support 29. In this way, both the radial and the axial positions of the ring 18 are fixed by the support 29. The axial length of the section 56 is established so that the forward surface 60 of the rear radial section 58 abuts the rear side 65 of the vane outer band 19 so as to provide axial support therefor. It will, therefore, be recognized that as the vane 11 is loaded, the axial load will be transmitted to the nozzle support 29 by way of the enlarged section 58, the axial section 56 and the rear radial section 59 of the ring 18. At the same time, since the rear surface 65 of the vane outer band 19 is allowed to slide over the forward surface 60 of the rear radial section 58, the tendency for the radial loads to be transferred from the vane to the ring, and hence to the support 29, are minimized.

As can be seen by reference to FIG. 2, a cavity 62 is generally defined by the ring 18, the nozzle support 29, and the shroud support segments 36. Airflow communication is provided between this cavity 62 and the cavity 24 (FIG. 1) by a plurality of holes 63 which are properly sized to meter the air which flows by way of the ports 53 and the plenum 54 to cool the shroud structure. In addition, this row of holes as can be seen in FIG. 3 acts to isolate the upstream end of the ring axial section 56 from the thermal variations in the downstream end thereof. That is, since the ring rear radial section 58 is partially exposed to the hot gas stream, it is subject to temperature variation and to nonuniform circumferential temperature gradients. However, the row of holes 63 tends to form a barrier which prevents these temperatures from migrating to the enlarged section 59 of the ring and, hence, to the nozzle support 29 to cause nonuniformity in the circumference of the nozzle support 29 and of the shroud segments 33. It should be noted that the location of the holes 63 could be moved upstream or downstream on the ring, or could even be located in the nozzle support 29 itself.

Referring now to FIGS. 4 and 5, the C-clamp inner flange 47 is shown to be in a close-fit relationship with the shroud rear groove 48 so as to thereby assist in clamping the shroud segment 33 to the rim inner flange 42 so as to accurately control the radial position of the shroud 33. This close-fitting, interlocking relationship between the shroud and the rim inner flange 47 further prevents the hot flow path gases from flowing radially outward into the cavity 64 defined by the C-clamp 43 and the rim 39. Therefore, the cavity 64, the rim 39 and the support 29, as well as the adjacent rim outer flange 41, remain at a uniform cool temperature to maintain the vane and shroud in a substantially stable radial position. Since the C-clamp inner flange 47 is positioned close to the main gas stream, the transition cavities are essentially eliminated and hot gas recirculation is prevented so as to retain the motive energy within the hot gas flow path.

As can be seen by reference to FIG. 4, there is an area between the shroud segment 33 and the vane outer band 44 where the hot gas leaving the rotating stage is susceptible to stagnating against the leading edge of the nozzle. However, in order to prevent this, the air which is used to cool the shroud is further used to prevent or reduce this hot gas recirculation. A plurality of axially extending, circumferentially spaced holes 66 are formed in the shroud so as to fluidly connect the cavity 52 to the shroud rear groove 48. Further, a plurality of axially extending, circumferentially spaced slots 67 are formed in the inner edge of the C-clamp inner flange 47 so as to provide fluid communication between the shroud rear groove 48 and the leading edges of the stage 2 vanes. This structure allows the cooling air from the outer side of the shroud to flow through the holes 66, circumferen-

tially around the shroud rear groove 48, through the slots 67, and into the area where the hot gas recirculation would otherwise occur. The cooling air then flows around the outer band leading edges to provide additional hot spot cooling and then re-enters the gas stream to do additional useful work. Proper alignment of the slots 67 with that region susceptible to airfoil stagnation, tends to flush out those regions and thereby maintain uniform temperatures and radial positions of the shroud and support structures.

Although the invention has been shown and described in terms of a preferred embodiment, it should be understood by those skilled in the art that various changes and omissions in the form and detail may be made therein without departing from the true spirit and scope of the invention.

Having thus described the invention, what is considered novel and desired to be secured by Letters Patent of the United States is:

1. In a turbomachine support structure of the type extending radially inward to provide radial support for the forward end of a turbine shroud and axial support for a vane outer band, an improved support arrangement comprising:

(a) a shroud for placement in close radial relationship with a row of turbine blades;

(b) a shroud support segment interconnecting said shroud and a surrounding support element, said support element having a forward flange extending independently forward from the point where said shroud support segment connects to said shroud support element;

(c) a ring disposed between the vane outer band and said support element, said ring comprising an axial leg with its front end abutting said forward flange and a radial leg integrally attached to and extending radially inward from the rear end of said axial leg to frictionally engage and provide axial support to the vane outer band; and

(d) means for introducing cooling air in the space partially defined by said ring and said shroud segment wherein said means comprises a plurality of circumferentially spaced holes formed in said ring for conducting the flow of cooling air therethrough.

2. The support structure as set forth in claim 1 and including means for supplying cooling air to the outer surface of said shroud.

3. A turbomachine support structure for jointly providing radial support to a turbine shroud and axial support to a turbine vane disposed forward of the shroud comprising:

(a) a support element disposed radially outward from both the shroud and the vane;

(b) a relatively rigid support segment interconnecting the shroud and the support element; and

(c) a relatively flexible link interconnecting the turbine vane to said support element, such that radial movement of the vane will be accommodated by a flexing of said relatively flexible link rather than by movement of said support element, said flexible link having a plurality of holes formed therein for the passing of cooling air therethrough for cooling a portion thereof.

4. A turbomachine support structure as set forth in claim 3 wherein said flexible link includes a substantially axially extending portion.

5. A turbomachine support structure as set forth in claim 4 wherein said substantially axially extending portion comprises a ring having a radially extending leg attached to the rear end thereof.

6. A turbomachine support structure as set forth in claim 3 wherein said flexible link includes an integral extension of said support element.

7. The support structure, as set forth in claim 1, wherein said plurality of holes are formed in said axial leg.

8. The support structure, as set forth in claim 7, wherein said plurality of holes are formed near the midpoint of said axial leg.

* * * * *

United States Patent [19]

Steel et al.

[11] **4,317,646**

[45] **Mar. 2, 1982**

[54] **GAS TURBINE ENGINES**

[75] Inventors: **Thomas Steel**, Littleover; **Noel H. Hooke**, Etwall; **George Pask**, Stanton-by-Bridge, all of England

[73] Assignee: **Rolls-Royce Limited**, London, England

[21] Appl. No.: **123,777**

[22] Filed: **Feb. 22, 1980**

[30] **Foreign Application Priority Data**

Apr. 26, 1979 [GB] United Kingdom 14606/79

[51] **Int. Cl.**^{3} **F01D 11/08**

[52] **U.S. Cl.** .... **415/116**; 415/117; 415/134; 415/172 A; 415/178

[58] **Field of Search** .... 415/116, 117, 178, 172 A, 415/134

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,863,634 12/1958 Chamberlain 415/134
3,583,824 6/1971 Smuland 415/134
3,825,364 7/1974 Halila 415/116
3,864,056 2/1975 Gabriel 415/116
4,251,185 2/1981 Karstensen 415/136

FOREIGN PATENT DOCUMENTS

2033021 5/1980 United Kingdom 415/117

Primary Examiner—Douglas Hart
Attorney, Agent, or Firm—Cushman, Darby & Cushman

[57] **ABSTRACT**

A device for controlling the clearance between the blade tips of a gas turbine engine turbine rotor and associated segmented shrouds comprises a control ring which is secured to the engine casing by a plurality of dowels. The control ring is supplied with cooling fluid to its hollow interior and is covered with a thermally insulating layer, whereby the rate of movement of the shroud segments can be maintained substantially equal to that of the adjacent turbine rotor.

6 Claims, 3 Drawing Figures

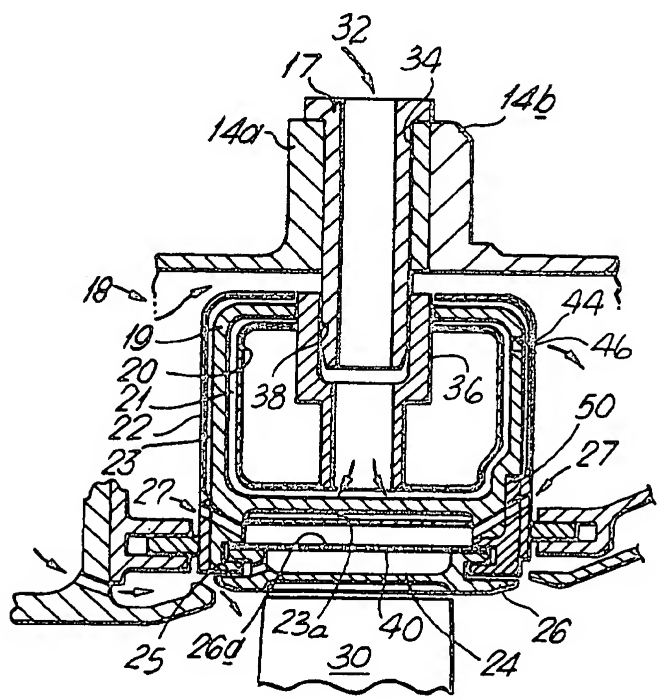


Fig. 1.

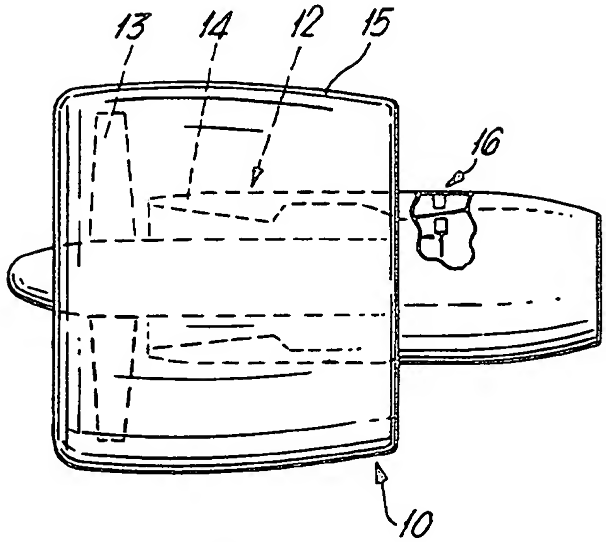

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Fig. 2.

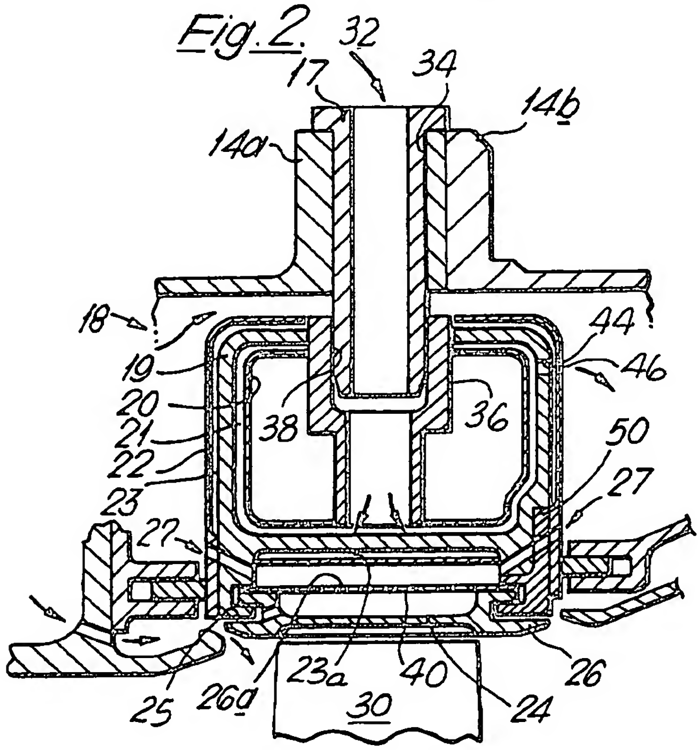
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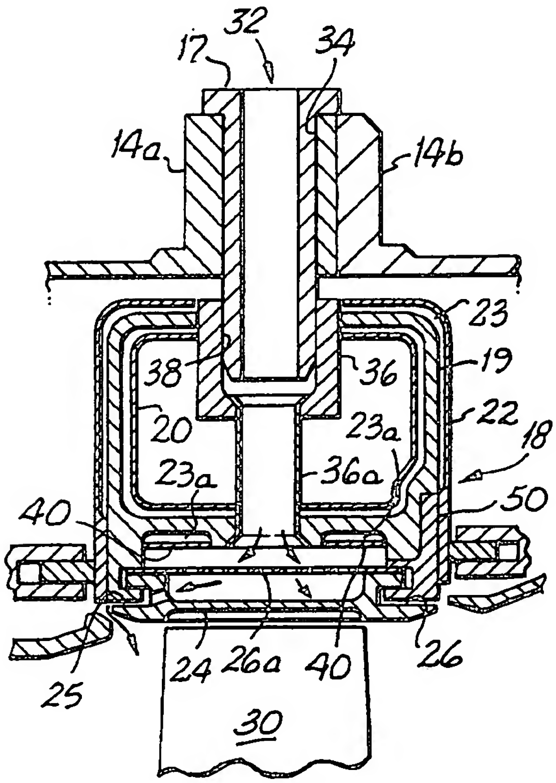
Fig. 3.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GAS TURBINE ENGINES

This invention relates to gas turbine engines and more particularly to such engines having a turbine rotor of the unshrouded type.

It is well known that in order for an unshrouded type turbine to operate efficiently the clearance between the turbine blade tips and the adjacent casing structure must be maintained within closely defined limits. The difficulties involved in maintaining such clearances have been well known for many years, and the problem has in fact become worse as both the size and working temperatures of gas turbine engines has increased.

One of the main factors which must be taken into account when designing a satisfactory arrangement is matching the respective diameters of the casing and the turbine at the different temperatures encountered during the working cycle of the engine, account must be taken of the differing coefficients of expansion of the materials involved together with the differing stresses imposed upon them, together with their different thermal response rate.

It must also be appreciated that while striving to maintain the smallest possible radial clearance between the respective components the design must be such as to avoid any interference occuring between the respective components during the differing working cycles to which the engine is subjected.

An object of the present invention is to provide a device for controlling the blade radial tip clearance which substantially eliminates the aforementioned problems.

According to the present invention a device for controlling the clearance between the blade tips of a gas turbine rotor and its associated casing structure comprises a hollow annular control ring to which is secured a plurality of shroud segments which surround the blade tips and define the clearance therebetween, the control ring being secured to the engine casing by a plurality of radially extending hollow dowels which are adapted to supply fluid into the interior of the control ring, the ring being covered with a thermally insulating layer whereby the rate of radial movement of the segments can be maintained substantially equal to that of the adjacent bladed turbine rotor which they surround during at least part of the engine operating cycle.

According to a further aspect of the invention the control ring is secured to the outer engine casing by dowels such as to substantially isolate the control ring from any deformation or expansion occuring within the casing.

Furthermore the fluid flow comprises high pressure air bled from the compressor section of the engine, which air is also used to cool the segmented shroud.

Preferably the fluid flow provided to the interior of the control ring may be used to assist in the control of the expansion and contraction of the ring to thereby control the clearance between the segments and the blade tips.

Preferably the thermally insulating layer applied to the control ring comprises a metal foil which defines an insulating air space between the foil and the control ring, alternatively or in addition the insulating layer comprises a refractory material such as for example magnesia stabilized zirconia.

For better understanding of the invention an embodiment thereof will now be more particularly described by way of example only, and as illustrated in the accompanying drawings in which:

FIG. 1 shows a pictorial view of a ducted fan type gas turbine engine having a broken away portion of its turbine casing disclosing a diagrammatic view of an embodiment of the present invention,

FIG. 2 shows a more detailed cross-sectional view of an enlarged scale of the embodiment shown diagrammatically at FIG. 1.

FIG. 3 shows a cross-sectional view of a further embodiment of the present invention.

Referring to FIG. 1 of the drawings a ducted fan type gas turbine engine shown generally at **10** includes a main core engine shown generally at **12** which serves to drive a front fan **13** situated within a fan duct which is defined by a portion of the core engine casing **14** and the fan cowling **15**. A portion of the core engine casing surrounding the turbine section of the engine shown generally at **16** is broken away, to show a diagrammatic view of one embodiment of the present invention.

FIG. 2 of the drawings shows a cross-sectional view in greater detail of the embodiment shown diagrammatically at FIG. 1 and consists of two engine casing portions **14***a* and **14***b* each of which terminate in abutting flanges which are secured together by a plurality of axially extending bolts not shown in the drawings.

As will be seen from the drawings the flange on casing portion **14***a* includes locally thickened portions in which are provided with a plurality of circumferentially spaced apart radially extending drillings **34** within each of which is located a hollow dowel one of which is shown at **17**. The hollow dowels **17** serve to carry a hollow annular control ring shown generally at **18** which is provided with circumferentially spaced bosses **36** having radial drillings **38** therethrough which correspond with the drillings **34** provided within the flange of casing portion **14***a*. The control ring **18** is located or secured by means of the dowels **17** such that control ring **18** has the ability to expand and contract independently of the casing portions **14***a* and **14***b* and their associated flanges.

The control ring shown generally at **18** comprises a main hollow annular member **19** which is made such that its thermal rate of expansion and contraction closely matches those of the turbine rotor of which one blade **30** is shown. A further annular member **20** is provided within the annular member **19** and is located such as to define a space **21** between the two members **19** and **20** which may be provided with a supply of air through the hollow dowels **17**. A metal foil **22** is located over and secured to but can move radially independently of the hollow annular member **19** and is arranged such as to define a space **23** between the two members **19** and **22**. A similar space **23***a* is defined between member **19** and a cylindrical member **40**. These spaces are filled with air and are sealed such that the air acts as heat insulating material upon the exterior of the annular member **19**. However it is envisaged that the space **23** could be filled with some other suitable insulating material, for example asbestos. Alternatively member **22** and space **23** could be entirely eliminated and be replaced by a layer or layers of some suitable ceramic refractory material such as for example magnesium zirconate. Alternatively the space **23** may be utilized to carry the supply of high pressure air from the dowels **17**.

Provided radially inwardly of the hollow annular ring member **19** are the segmented shroud portions, one of which is shown at **24**. Each shroud portion is pro-

vided with axially extending recesses **25** and **26** situated one on either end of the segmental shroud portions **24** by means of which the shroud portions may be secured to the hollow annular ring member **19** by a cooperating portion **42** provided upon the radially innermost end of the annular member **19** and by member **50** which is secured to annular member **19**. In this way movement of the shroud segments **24** is limited to the movements of the hollow member **19**.

Situated within the annular space defined by the shroud segments **24** and the radially innermost portion of the hollow member **19** is a perforate cylindrical member **26a** which is provided with a supply of cooling air **27** which passes through the perforate member **26a** and impinges upon the shroud segments such as to provide them with a degree of cooling. The cooling air **27** is obtained from some suitable location within the compressor section of the core engine.

As previously stated during the operating cycle of a gas turbine engine its components are subjected to changes in both temperature and mechanical stress which changes the tip clearance between the turbine blades **30** and the adjacent shroud segments **24**. However, it is believed that by carefully controlling the degree of expansion or contraction to which the ring **19** is subjected, it is possible to maintain the tip clearance within reasonable limits, during critical parts of the engine cycle.

Obviously the engine is designed such that its blade tip clearances are at their minimum size when it is in the cruise condition. However it is also important that the engine is running as efficiently as possible particularly during the 'take-off' and 'climb' engine conditions.

On engine 'start-up' and subsequent 'ground-idle' the turbine blades **30** tend to expand relatively quickly due to both thermal expansion and centrifugal loading, and this tends to close up the radial clearance between the blade tips and the shroud segments **24**; however such reduction in clearance can be simply allowed for in the design of the engines by suitable sizing of the relative parts, as it is unimportant if there is an excessively large clearance when the engine is cold and stationary, or during taxi or descent.

When the engine is accelerated to its 'high power' condition for 'take off' and 'climb' the turbine blades undergo a further expansion which is caused by both centrifugal force, and thermal expansion. However expansion also occurs in the turbine discs which increase in diameter, due to both thermal expansion and centrifugal force. During this time the shrouds and casing undergo some expansion due to thermal effects and pressure. However such expansion would not prove to be so sufficient to maintain an adequate tip clearance without the provision of the hollow annular control ring structure **18**.

Matching the respective growth rates of the turbine shrouds **24** to that of the turbine blades so as to maintain an acceptable tip clearance during the 'take-off' and 'climb' mode of the engine cycle is achieved by matching the rate of expansion of the control ring **19** to that of the turbine rotor disc and blade structure.

Such matching is achieved by providing the hollow annular ring member **19** with a thermally insulating barrier comprising the annular member **22** which retains an insulating layer of air in the spaces **23** and **23a** around the member **19** such that it remains partially isolated from its environment such that its rate of expansion can be matched so as to become similar to that encountered by both the turbine disc and turbine blades during this part of the engine flight cycle. However the rate of expansion or contraction of the hollow annular control ring member **19** may be further controlled by means of a supply of high pressure air **32** which is bled from the compressor section of the engine into the member **19** through the hollow radially extending dowels **17**. The high pressure air passes through space **21** and is subsequently exhausted through vents **44** and **46** in members **19** and **22** respectively. Alternatively the high pressure air passes through space **23** and is subsequently exhausted through vents **46** in member **22**.

FIG. 3 shows a further embodiment of the present invention, however in this case the air supply **32** is also used to provide impingement cooling direct to the shroud segments **24** after passing through the control ring structure shown generally at **18**. In more detail, the bosses **36** are provided with extensions **36a** which discharge the air **32** directly into the area above the perforate cylindrical member **26a**. Air passing through the member **26a** impinges directly onto the shroud portions **24**.

We claim:

1. In a gas turbine engine, the improvement in structure for controlling clearance between blade tips of a turbine rotor having a predetermined rate of expansion and shroud means surrounding the blade tips, the combination comprising:

an engine casing;

a hollow annular control ring concentrically positioned within said casing;

a plurality of shroud segments defining said shroud means about the blade tips of the turbine rotor, said shroud segments being secured to said hollow annular control ring for expansion and contraction with the same;

means operatively supporting said hollow annular control ring and the shroud segments supported thereby from said casing while mechanically isolating said control ring from deformation or expansion of said casing, said support means permitting said control ring to expand and contract independently of said casing while being maintained concentric with the same, and said support means providing means to supply a fluid to the interior of said hollow annular control ring, and means thermally insulating the exterior of said hollow annular control ring, said thermally insulating means and said support means cooperating to control the rate of expansion of said control ring to match the rate of expansion of the turbine rotor whereby clearance between the shroud means and the blade tips of the turbine rotor is a predetermined amount during at least a portion of the operating cycle of the engine.

2. A gas turbine engine as claimed in claim **1** in which the fluid flow comprises high pressure air bled from the compressor section of the engine, which air is also used to cool the shroud segments.

3. A gas turbine engine as claimed in claim **1** in which the thermally insulating means for the exterior of the control ring comprises a metal foil which defines an insulating air space between the foil and the control ring.

4. A gas turbine engine as claimed in claim **1** in which the thermally insulating means for the exterior of the control ring comprises a refractory material extending about the exterior of said control ring.

5. A gas turbine engine as claimed in claim 4 in which the refractory material comprises magnesia stabilized zirconia.

6. A gas turbine engine as claimed in any one of claims 1, 2, 3, 4 or 5 in which said support means for said control ring includes a plurality of circumferentially spaced hollow dowels extending radially through said casing and received in a plurality of radially extending drillings in said hollow annular control ring, said supply fluid to the interior of said control means being supplied through said hollow dowels.

* * * * *

United States Patent [19] | [11] **4,063,845**
Allen | [45] **Dec. 20, 1977**

[54] TURBOMACHINE STATOR INTERSTAGE SEAL

[75] Inventor: Robert L. Allen, Greenwood, Ind.
[73] Assignee: General Motors Corporation, Detroit, Mich.

[21] Appl. No.: 732,285
[22] Filed: Oct. 13, 1976

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 583,548, June 4, 1975, abandoned.

[51] Int. Cl.[2] F01D 25/26; F01D 9/00; F04D 29/40; F16J 15/48
[52] U.S. Cl. .... 415/134; 415/217; 415/219 R; 277/204; 277/236
[58] Field of Search .... 415/134–139, 415/171, 172 A, 173 R, 216–218, 219 R, 196; 277/204, 236

[56] **References Cited**

U.S. PATENT DOCUMENTS

843,740	2/1907	Fenn	277/204
1,543,963	6/1925	Walton	277/204
2,819,919	1/1958	Pearce et al.	277/204
2,991,045	7/1961	Tassoni	415/135
3,004,700	10/1961	Warren	415/216
3,132,870	5/1964	Pschera	277/204
3,728,041	4/1973	Bertelson	415/218
3,754,766	8/1973	Asplund	277/236
3,869,222	3/1975	Rahnke et al.	415/134

FOREIGN PATENT DOCUMENTS

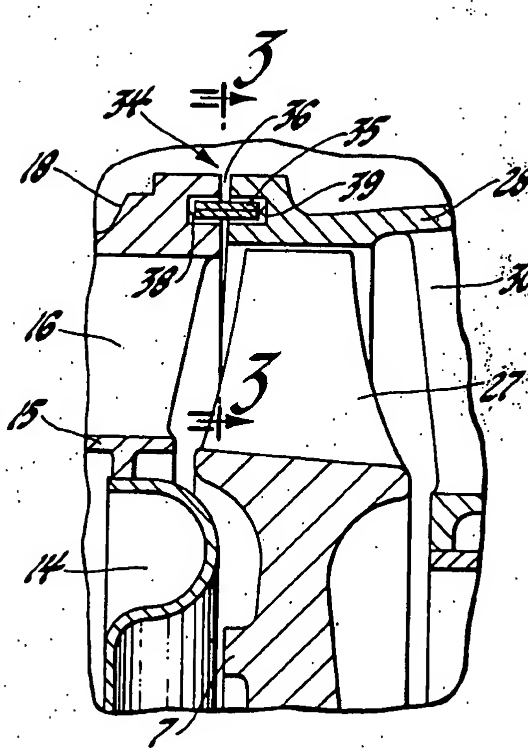
244,726	2/1927	United Kingdom	415/136
180,822	6/1922	United Kingdom	277/204

Primary Examiner—John J. Vrablik
Attorney, Agent, or Firm—J. C. Evans

[57] **ABSTRACT**

Successive outer stator shroud rings of a gas turbine confront each other at the downstream edge of one and the upstream edge of the other, with a gap between them. To seal against leakage through this gap, edges of the shrouds have grooves aligned to define an annular cavity of generally rectangular cross section in the shrouds. A thin flexible coiled metal strip of about 720° extent is disposed in the cavity, with the strip overlapping itself to form a double layer. Layers of the strip are slidable relative to each other to allow the strip to expand and contract with the shrouds and abut outer or inner walls of the cavity to provide the seal.

3 Claims, 8 Drawing Figures



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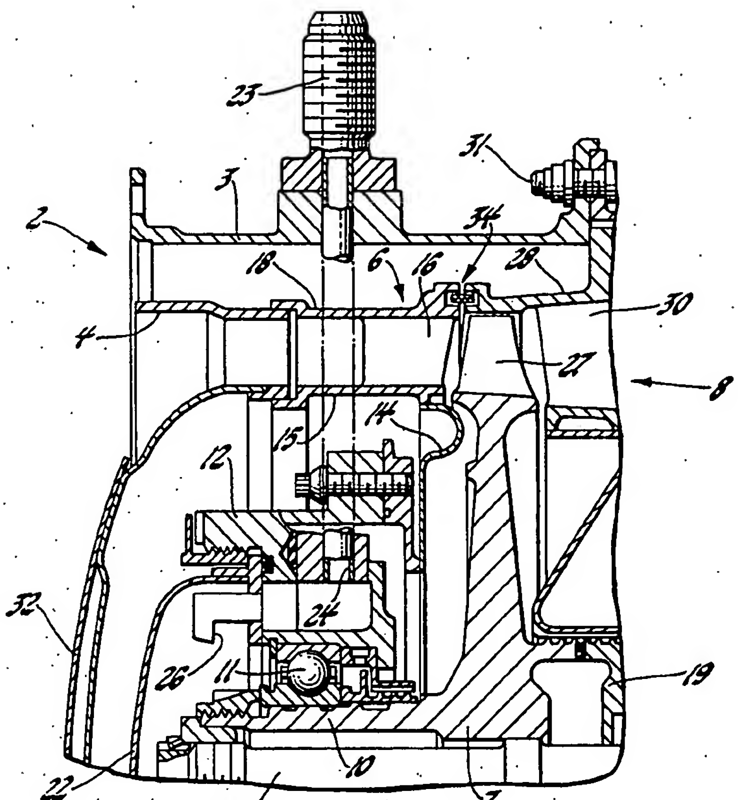

Fig. 1

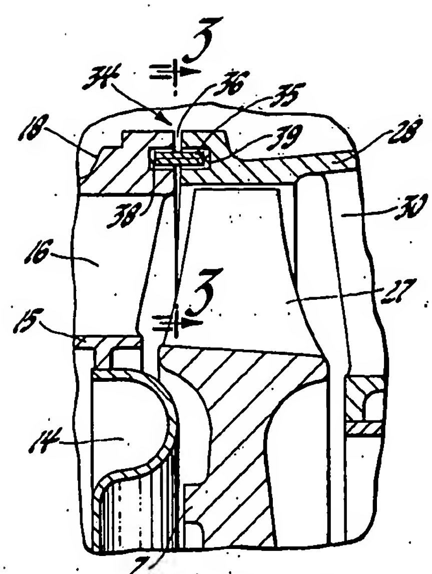

Fig. 2

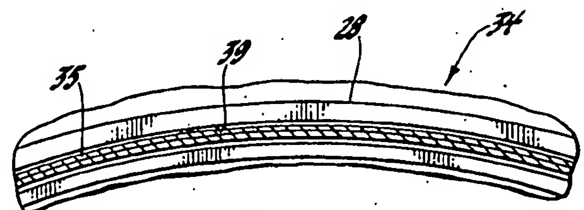

Fig. 3

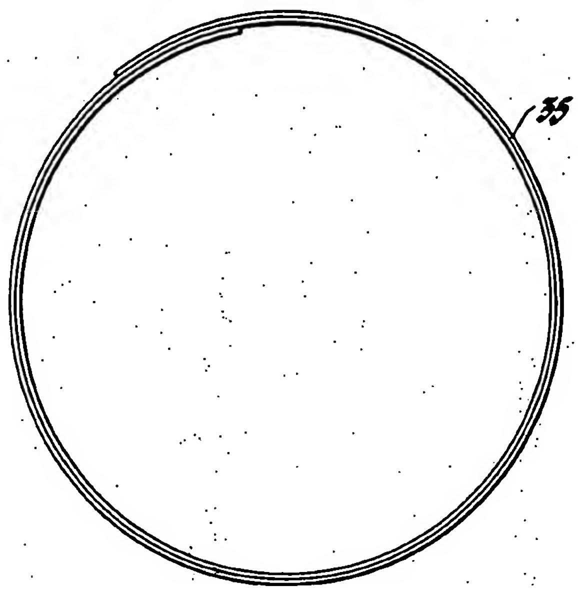

Fig. 4

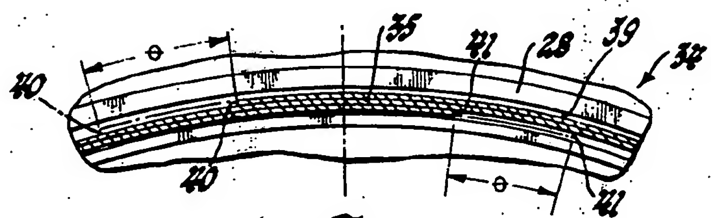

Fig. 5

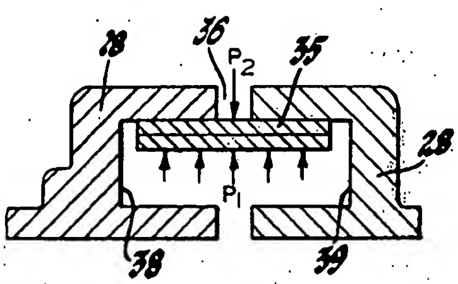

Fig. 6

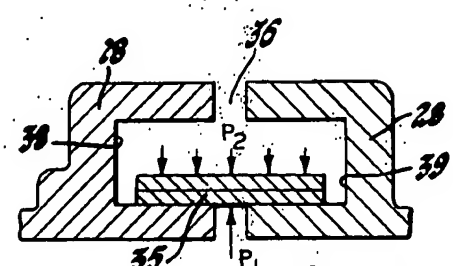

Fig. 7

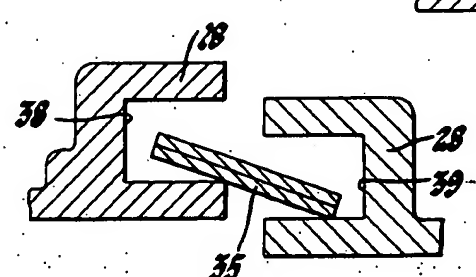

Fig. 8

TURBOMACHINE STATOR INTERSTAGE SEAL

This is a c ntinuation-in-part of Ser. No. 583,548 filed June 4, 1975 now abandoned.

My invention is directed to improved stator structures for turbomachines, and particularly to a new and improved seal to minimize leakage through a gap between successive shrouds of a turbine stator. In axial-flow turbines such as are used in gas turbine engines, for example, the outer boundary of an annular motive fluid path through the turbine is provided by stationary shrouds or stator shrouds. Such a shroud may be integral with or connected to the stator vanes which extend from the outer shroud to an inner shroud and direct the motive fluid into a turbine rotor stage. Or, a shroud may lie immediately outward of a rotor stage of the turbine so that the motive fluid is constrained to flow through the blades of the rotor stage.

For various structural reasons, the turbine stator shrouds are ordinarily constituted by successive sections from inlet to outlet of the turbine. Ordinarily some clearance must be provided between successive shrouds to allow for expansion, or for other reasons. There may be a pressure difference between the motive fluid inside the shroud and the air or other gas external to the shroud. The air external to the shroud may be air tapped from the compressor of the engine and ordinarily is at a higher pressure than the motive fluid. In any event, it is usually undesirable to have uncontrolled leakage through a gap between successive outer shrouds of a turbine.

Various arrangements for sealing such gaps have been proposed, but so far as I am aware, none similar to the one which is the primary subject matter of this application. According to my invention, a very simple sealing arrangement is provided by a flexible coil of thin sheet metal strip lying in opposed grooves in confronting faces of the shroud ring. The metal strip preferably is of about 720° extent so that a two layer coil of the strip is provided. Under the action of pressure differences and vibration in the engine, this coil can adjust itself inwardly or outwardly to lodge against the outer or inner walls of the grooves in the shroud ring, depending upon the direction of the pressure gradient.

The principal object of my invention is to provide a simple, durable, and inexpensive seal between successive shroud rings of a turbine stator.

The nature of my invention and its advantages will be clear to those skilled in the art from the succeeding detailed description of the preferred embodiment of the invention and the accompanying drawings thereof.

FIG. 1 is a partial view of a gas turbine taken on a plane containing the axis of rotation of the turbine,

FIG. 2 is an enlargement of a portion of FIG. 1,

FIG. 3 is a fragmentary cross-sectional view taken on the plane indicated by the line 3—3 in FIG. 2,

FIG. 4 is an expanded end elevational view of a metal coil strip in the present invention,

FIG. 5 is a fragmentary sectional view to illustrate movement of end segments of the coil in FIG. 4 to accommodate changes in coil diameter,

FIG. 6 is an enlarged fragmentary sectional view of a first operating position of the coil strip and shroud components,

FIG. 7 is a view like FIG. 6 showing a second operating position, and

FIG. 8 is a view like FIG. 6 showing yet another operating position.

Referring first to FIG. 1, the seal structure of the invention is shown as incorporated in a gas turbine aircraft engine of kn wn type having the United States military designation T63. The figure shows the first stage and part of the second stage of the high pressure turbine 2 of such an engine. The turbine 2 comprises an outer case ring 3, a turbine inlet annulus 4, a first stage turbine nozzle 6, a first stage turbine wheel 7, and a second stage turbine nozzle 8. The turbine wheel 7 is integral with a stub shaft 10 rotatably mounted by a ball bearing 11 in a bearing mount 12. The bearing mount is supported from the outer case ring 3 by struts (not illustrated). The nozzle 6 is supported from the bearing mount 12 by a flexible diaphragm 14 (see also FIG. 2) bolted to the bearing mount. The diaphragm is welded to the inner shroud 15 of the turbine nozzle 6, which is connected integrally by nozzle vanes 16 to an outer shroud ring 18. In this engine, the shroud 18 is a continuous integral 360° ring. The invention is applicable also to segmented shrouds, however.

The turbine wheel 7 is coupled to a second stage wheel 19, shown fragmentarily, by a tie bolt 20. Bearing 11 is disposed in a housing defined in part by a cover 22 which provides an oil sump for the bearing. Oil is supplied to the bearing from a connection 23 through an oil line 24 and a jet 26.

The first stage turbine wheel 7 bears a ring of blades 27 which are driven by the gas discharged from the nozzle 6. These vanes rotate closely within a second stage shroud 28 which is integral with vanes 30 of the second stage turbine nozzle. Shroud 28 is fixed through an expansion joint to the outer case ring 3 by bolts 31.

The engine combustion apparatus has an annular wall the outlet end of which is piloted within the turbine inlet annulus 4. This outlet is defined in part by a shield 32 disposed ahead of the turbine.

The foregoing constitutes a sufficient description of a presently preferred environment for my invention.

Referring now to FIGS. 2, 3 and 4 for a description of the seal arrangement 34, the seal is defined by a 720° coil 35 of thin strip metal, preferably in this case of 0.025 mm. thickness, which bridges the gap 36 between the shrouds 18 and 28. The seal strip is disposed in a groove 38 in the downstream edge of shroud 18 and a confronting groove 39 in the upstream edge of shroud 28. These grooves preferably are formed by machining. The strip has some degree of clearance axially and also radially so that it is free to expand and contract and expand differentially to the shrouds as temperature changes. Preferably, with a strip 0.025 mm. thick the radial dimension of the grooves 38 and 39 is 0.075 mm., leaving 0.025 mm. clearance in the radial direction. There is nothing critical about the axial clearance.

The coil 35 is preferably made of any high temperature resisting metal sheet or strip. Examples of suitable materials include AMS 5759 (a cobalt base high temperature material) and AMS 5545 (a nickel base high temperature material). Because of its small thickness, it is quite flexible and has an inherent bias that causes it to accommodate itself to lie against the inner or outer wall of the grooves 38 and 39. The coil 35 does not take a permanent set and it eliminates the need for a separate spring element to seat the seal coil. Assuming that the direction of pressure P_2 is from outside to inside, P_2 being greater than P_1, the pressure differential exerted through gap 36 will seat the ring against the inner wall of the grooves as shown in FIG. 7. The strip can slip circumferentially in response to gas pressure or physical

force exerted by the supporting shrouds. The ability to slip is aided by the slight vibration due to rotation of the turbine. The circumferential slip action is shown in FIG. 5 wherein ends **40**, **41** on the coil **35** are shown shifted by an angle θ, illustrating movement produced under increased pressure P_2. Conversely, when pressure P_1 exceeds P_2 the coil **35** expands circumferentially by slip between segments of the coil **35** and will seat against the outer wall of the grooves as shown in FIG. 6.

The twice wound coil **35** and open face configuration of grooves **38**, **39** has general application in turbine stator structures having radial pressure differentials to produce the effects shown in FIGS. 6 and 7. Further, in some high temperature turbines, the shroud ring **18** will be maintained at a higher temperature than the downstream second stage shroud **28**. Thus, shroud ring **18** will have a greater diameter than shroud **28** because of thermal difference. The coil **35** slides and cants to accommodate this difference as shown in FIG. 8.

The simple and adequate character of the seal arrangement should be clear to those skilled in the art from the foregoing description.

The detailed description of the preferred embodiment of the invention for the purpose of explaining the principles thereof is not to be considered as limiting or restricting the invention, since many modifications may be made by the exercise of skill in the art.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

1. A turbine stator comprising, in combination, an annular upstream shroud having a continuously circumferentially formed downstream edge; an annular downstream shroud coaxial with the upstream shroud having a continuously circumferentially formed upstream edge confronting the downstream edge of the upstream shroud; said downstream edge being axially spaced from said upstream edge throughout the full circumferential extent of both said downstream and upstream edges to form an open gap extending radially therebetween continuously circumferentially therebetween; the confronting edges of the shrouds defining opposed grooves in the confronting edges; and seal ring means operable to oppose gas flow through the gap between the shrouds, the seal ring means being a thin flexible coiled metallic strip extending substantially twice around the axis of the shrouds and axially across the gap and within the grooves to provide two layers, the layers being mutually slidable to accommodate relative radial expansion of the shrouds and strip for seating of a surface of the strip radially against the shrouds.

2. A turbine stator comprising, in combination, an annular upstream shroud having a continuously circumferentially formed downstream edge; an annular downstream shroud coaxial with the upstream shroud having a continuously circumferentially formed upstream edge confronting the downstream edge of the upstream shroud; said downstream edge being axially spaced from said upstream edge throughout the full circumferential extent of both said downstream and upstream edges to form an open gap extending radially therebetween continuously circumferentially therebetween; the confronting edges of the shrouds defining opposed grooves in the confronting edges; and seal ring means operable to oppose gas flow through the gap between the shrouds, the seal ring means being a thin flexible coiled metallic strip extending substantially twice around the axis of the shrouds and axially across the gap and within the grooves to provide two layers, the layers being mutually slidable in response to pressure difference across the gap to accommodate relative radial expansion of the shrouds and strip for seating of a surface of the strip radially against the shrouds.

3. A turbine stator comprising, in combination, an annular upstream shroud having a continuously circumferentially formed downstream edge; an annular downstream shroud coaxial with the upstream shroud having an upstream edge confronting the downstream edge of the upstream shroud; the confronting edges of the shrouds defining opposed grooves in the confronting edges; and seal ring means operable to oppose gas flow through a gap between the shrouds, the seal ring means being a thin flexible coiled metallic strip extending substantially twice around the axis of the shrouds across the gap and within the grooves to provide two layers, the layers being mutually slidable to accommodate relative expansion of the shrouds and strip for seating of a surface of the strip radially against the shrouds, the strip having a thickness of the order of 1/40 millimeter.

* * * * *

United States Patent [19]

North

[11] **Patent Number: 4,902,198**

[45] **Date of Patent: Feb. 20, 1990**

[54] **APPARATUS FOR FILM COOLING OF TURBINE VAN SHROUDS**

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[21] Appl. No.: **238,942**

[22] Filed: **Aug. 31, 1988**

[51] **Int. Cl.**^{4} **F01D 5/18**

[52] **U.S. Cl.** **415/115**; 415/116

[58] **Field of Search** .... 415/115, 116, 170 R, 415/134, 138

[56] **References Cited**

U.S. PATENT DOCUMENTS

3,519,366	7/1970	Campbell	415/138
3,542,483	11/1970	Gagliardi	415/136
3,752,598	8/1973	Bowers et al.	415/115
3,938,906	2/1976	Michel et al.	415/139
3,947,145	3/1976	Michel et al.	415/217
3,975,114	8/1976	Kalkbrenner	415/217
4,650,394	3/1987	Weidner	415/115
4,688,988	8/1987	Olsen	415/116
4,720,236	1/1988	Stevens	415/116
4,767,260	8/1988	Clevenger et al.	415/115

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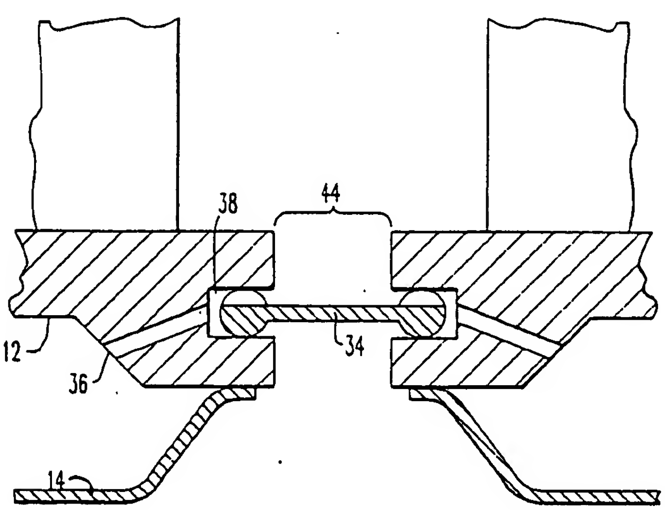
Assistant Examiner—John T. Kwon
Attorney, Agent, or Firm—K. Bach

[57] **ABSTRACT**

A gas turbine of the type having high pressure air supplied to the cavity formed by the inner shrouds of the turbine vanes is provided with film cooling of the shrouds. A manifold supplies high pressure cooling air to portions of the gaps between inner shrouds not otherwise supplied and intermittent reliefs in the strip seal between shrouds regulates the leakage of this air, over the outer surfaces of the shrouds.

14 Claims, 3 Drawing Sheets



1, 8, 9, 10

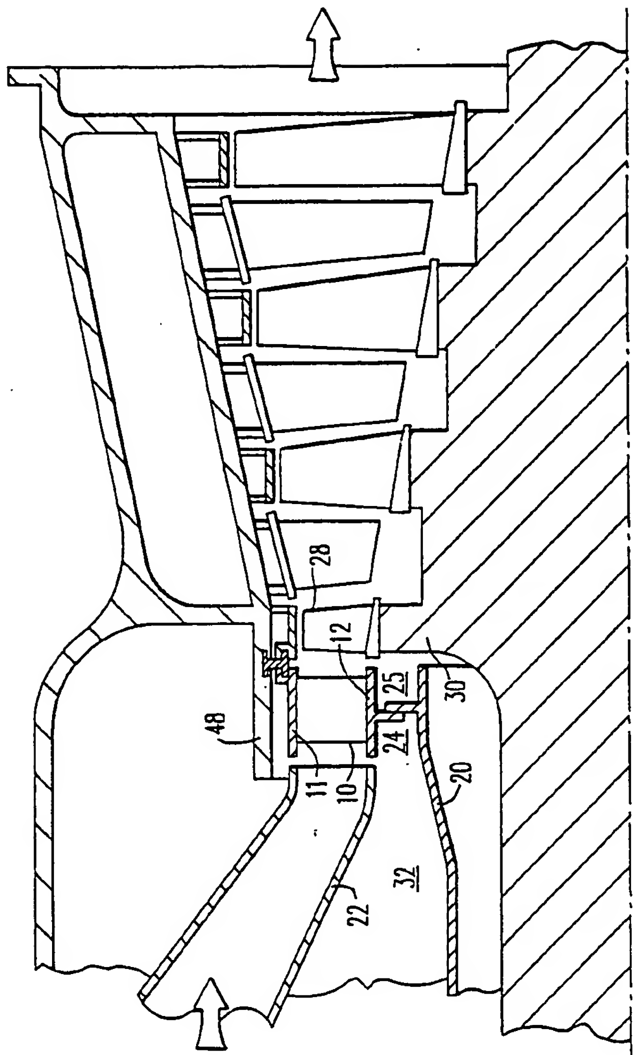


FIG. 1

FIG. 2

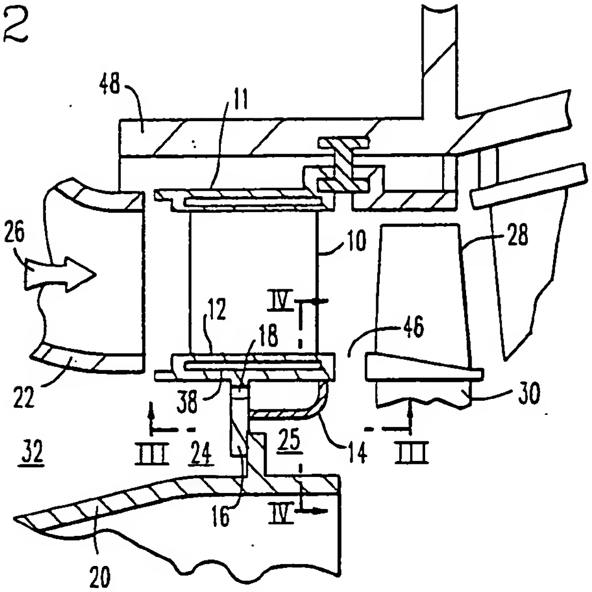
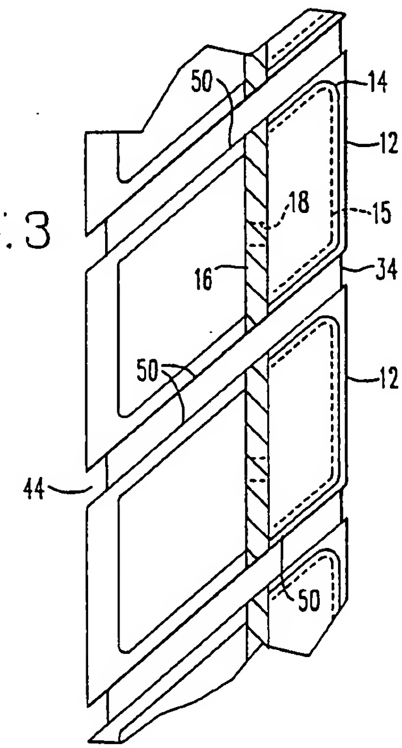
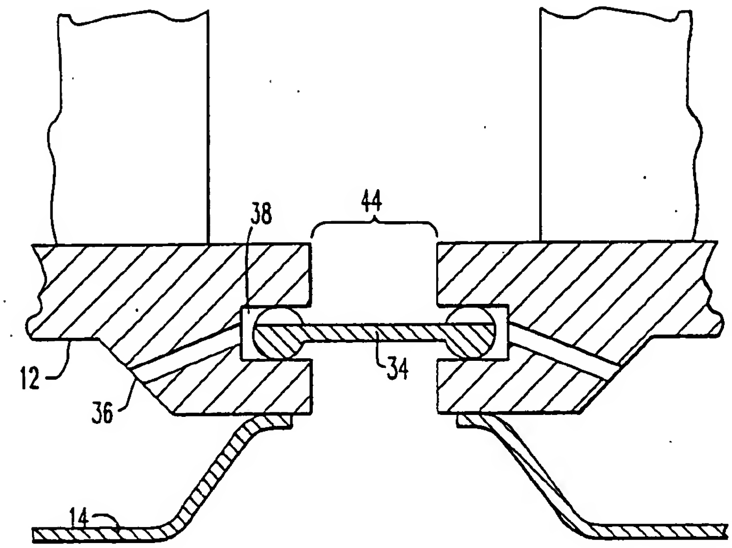


FIG. 3



vane to vane

FIG. 4



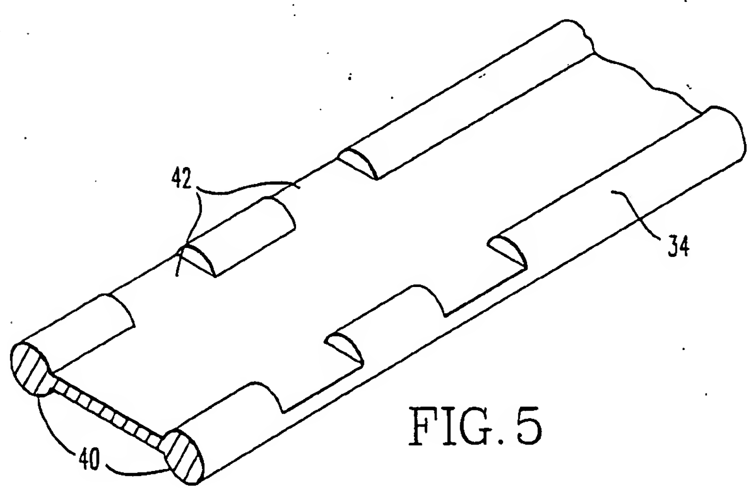


FIG. 5

APPARATUS FOR FILM COOLING OF TURBINE VAN SHROUDS

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention generally relates to gas turbines. More specifically, the present invention relates to an apparatus and method for supplying film cooling to the inner shrouds of the turbine vanes.

To achieve maximum power output of the turbine it is desirable to operate with as high a gas temperature as feasible. The gas temperatures of modern gas turbines are such that without sufficient cooling the metal temperature of the flow section components would exceed those allowable for adequate durability of the components. Hence, it is vital that adequate cooling air be supplied to such components. Since to be effective such cooling air must be pressurized, it is typically bled off of the compressor discharge airflow thus bypassing the combustion process. As a result, the work expended in compressing the cooling air is not recovered from the combustion and expansion processes. It is, therefore, desirable to minimize the use of cooling air to obtain maximum thermodynamic efficiency, and the effective use of cooling air is a key factor in the advancement of gas turbine technology. The present invention concerns the supply and control of film cooling air to the inner shrouds of the turbine vanes.

2. Description of the Prior Art

The hot gas flow path of the turbine section of a gas turbine is comprised of an annular chamber contained within a cylinder and surrounding a centrally disposed rotating shaft. Inside the annular chamber are alternating rows of stationary vanes and rotating blades. The vanes and blades in each row are arrayed circumferentially around the annulus. Each vane is comprised of an airfoil and inner and outer shrouds. The airfoil serves to properly direct the gas flow to the downstream rotating blades. The inner and outer shrouds of each vane nearly abut those of the adjacent vane so that, when combined over the entire row, the shrouds form a short axial section of the gas path annulus. However, there is a small circumferential gap between each shroud.

Generally high pressure air is present in the annular cavity formed by the inner surface of the inner shrouds. This is so in the first vane row because it serves as the entrance to the turbine section and hence is immediately connected to a plenum chamber containing compressor discharge air awaiting introduction into the combustion system. As a result of this arrangement high pressure compressor discharge air fills the cavity formed between the inner shrouds of the first row vanes and the outer surface of the housing which encases the shaft in this vicinity. In the vane rows downstream of the first row a somewhat different situation exists. To cool the rotating discs of the blade rows immediately upstream and downstream of the vane row, cooling air is supplied to the cavity formed by the inner shrouds and the faces of the adjacent discs.

Leakage of the high pressure air in these cavities into the hot gas flow results in a loss of thermodynamic performance. Hence means are employed to restrict such leakage. Since the pressure of the hot gas flow drops as it traverses downstream through each succeeding row in the turbine, the natural tendency of the high pressure air in these cavities is to leak out of the cavity by flowing downstream through the axial gap between the trailing edge of the inner shroud and the rim of the adjacent rotating disc. This is prevented by a radial barrier extending circumferentially around the annular cavity. In the first vane row this barrier comprises a support rail, emanating radially inward from the inner shroud inner surface, which serves to support the vane against the housing encasing the shaft. Although a hole may be provided in the support rail allowing high pressure air to flow across it, a containment cover affixed to the inner surface of the inner shroud prevents the high pressure air from entering the shroud cavity downstream of the barrier. In rows downstream of the first row, the barrier comprises a similar support rail to which is affixed an interstage seal.

A second potential leakage path of the high pressure air in the shroud cavity is through the circumferential gaps between adjacent inner shrouds. In the past such leakage has been prevented by strip seals disposed in slots in the edges of the inner shrouds forming the gaps. In earlier turbine designs leakage past these seals resulted in a thin film of cooling air flowing over the outer surface of the inner shroud. This film cooling was sufficient to prevent overheating of the inner shrouds. However, as advances in gas turbine technology allow increasingly higher hot gas temperatures, it may be anticipated that the leakage past the seals will become insufficient, especially in the portion of the shroud downstream of the radial barrier, where the pressure of the air, and hence the leakage rate, is lower. In such advanced turbines overheating can occur on the first vane row in the portion of the inner shroud downstream of the radial barrier if adequate cooling is not provided. Since overheating of the shroud will cause its deterioration through corrosion and cracking, it results in the need to replace the vanes more frequently, a situation which is costly and renders the turbine unavailable for use for substantial periods.

It is therefore desirable to provide an apparatus and method which will achieve adequate film cooling of the inner shrouds in areas, such as downstream of the radial barrier, where the pressure of the air within the shroud cavity is low.

SUMMARY OF THE INVENTION

Accordingly, it is a general object of the present invention to provide a method and apparatus for film cooling of the inner shrouds of a gas turbine.

More specifically, it is an object of the present invention to provide a method and apparatus for film cooling the portion of the inner shroud not supplied with high pressure cooling air by regulating the leakage of high pressure air through the gaps between adjacent shrouds.

It is another object of the invention to distribute high pressure cooling air to the strip seals disposed in the gaps between shrouds and to regulate the leakage of the air across such seals.

Briefly, these and other objects of the present invention are accomplished in a gas turbine with a plurality of vanes, each vane having an inner shroud. There is a small circumferential gap between adjacent vanes and strip seals are disposed in slots in the shrouds to prevent leakage of air through the gaps. High pressure air is supplied to a portion of the cavity formed by the inner shrouds and a radial barrier prevents the high pressure air from reaching the portion of the shroud cavity downstream of the barrier. A containment cover affixed to each inner shroud allows high pressure air to flow

through holes in the radial barrier to an opening in the inner shroud downstream of the barrier, so as to supply the vane airfoil with cooling air.

In accordance with one important aspect of the invention, a plurality of holes are provided extending from the slots retaining the strip seals to the portion of the inner surface of the shroud encompassed by the containment cover. Thus the containment cover serves to manifold high pressure air to these holes and thence the slots retaining the strip seals.

In accordance with another important aspect of the invention, the sealing surfaces of the strip seal are intermittently relieved to regulate the leakage of high pressure cooling air across the seals. This leakage provides film cooling to the inner shroud.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal cross-section of the turbine section of a gas turbine;

FIG. 2 shows a portion of the longitudinal cross-section of FIG. 1 in the vicinity of the first row vanes;

FIG. 3 is across-section taken through line 3—3 of FIG. 2 showing the inner shrouds of two adjacent vanes;

FIG. 4 is a cross-section of the inner shroud taken through line 4—4 of FIG. 2;

FIG. 5 is a perspective view of the strip seal.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to the drawings, wherein like numerals represent like elements, there is illustrated in FIG. 1 a longitudinal section of the turbine portion of a gas turbine, showing the turbine cylinder 48 in which are contained alternating rows of stationary vanes and rotating blades. The arrows indicate the flow of hot gas through the turbine. As shown, the first row vanes 10 form the inlet to the turbine. Also shown are portions of the chamber 32 containing the combustion system and the duct 22 which directs the flow of hot gas from the combustion system to the turbine inlet. FIG. 2 shows an enlarged view of a portion of the turbine section in the vicinity of the first row vanes 10. As illustrated, the invention applies preferably to providing cooling air to the first row of shrouds, but is applicable to the other rows as well. At the radially outboard end of each vane is an outer shroud 11 and at the inboard end is an inner shroud 12. Each inner shroud has two approximately axially oriented edges 50 and front and rear circumferentially oriented edges. A plurality of vanes 10 are arrayed circumferentially around the annular flow section of the turbine. The inner and outer shrouds of each vane nearly abut those of the adjacent vane so that, when combined over the entire row, the shrouds form a short axial section of the gas path annulus. However, there are small circumferential gaps 44 between the approximately axially oriented edges 50 of each inner shroud and the adjacent inner shrouds, as seen in FIG. 4. A housing 20 encases the rotating shaft in the vicinity of the first row vanes. Support rails 16 emanating radially inward from each inner shroud support the vane against this housing.

High pressure air from the discharge of the compressor flows within the chamber 32 prior to its introduction into the combustion system. This high pressure air flows freely into a shroud cavity 24 formed between the inner surface of inner shrouds 12 and the shaft housing 20. Rotating blades 28 are affixed to a rotating disc 30 adjacent to the vanes. A gap 46 is formed between the down stream edge of the shroud 12 and the face of the adjacent disc 30. The support rails 16 provide a radial barrier to leakage of the high pressure air downstream by preventing it from flowing through the shroud cavity 24 and into the hot gas flow through the gap 46.

Referring to FIGS. 2-5, it is seen that hot gas 26 from the combustion system flows over the outer surfaces of the inner shrouds. Leakage of the high pressure air into this hot gas flow through the gaps 44 between shrouds is prevented by means of strip seals 34 of dumbbell-shaped cross section shown in FIGS. 4 and 5. There is one strip seal for each gap, the seal spans the gap and is retained in the two slots along the edges of adjacent shrouds forming the gap. The cylindrical portions 40 of the dumbbell shape run along the two longitudinal edges of the seal and reside in the slots 38. Since the diameter of the cylindrical portions is only slightly smaller than the width of the slot they provide a sealing surface.

Holes 18 are provided in the support rail 16, one hole for each inner shroud. The holes extend from the front to the rear face of the rail and are equally spaced circumferentially around the rail. A containment cover 14 affixed to the inner surface of the inner shroud allows high pressure air to flow through these holes in the support rail and into the vane airfoil through an opening 15 in the inner shroud. The containment cover extends axially from the rear face of the support rail to near the rear circumferentially oriented edge of the shroud and circumferentially it approximately spans the two edges forming the gaps, as shown in FIG. 3.

The portion of the shroud cavity 25 downstream of the support rail 16 is not supplied with high pressure air from the compressor, as a result of being sealed off from chamber 32 by the support rail 16. Hence under the prior art approach very little cooling air can be expected to leak past the strip seal 34 to cool the portion of the inner shroud downstream of the support rail. In accordance with the present invention a means is provided for distributing high pressure air to the gap downstream of the support rail by providing a plurality of holes 36 extending from the slots 38 to the inner surface of the inner shroud encompassed by the containment cover 14 as shown in FIG. 4. These holes allow the containment cover to act as a manifold so that the holes 18 in the support rail 16 can supply high pressure air to the slots containing the seal 34. In accordance with another feature of the invention, a means is provided for regulating and distributing the leakage through the seal by providing intermittent reliefs 42 in the cylindrical portions 40 of the seal 34 downstream of the radial barrier, as shown in FIG. 5, the size and quantity of which determine the amount of leakage. The amount of leakage flow provided in this manner can also be controlled by varying the size of the holes 18 in the support rail 16. This leakage of high pressure air past the seals and through the circumferential gap between inner shrouds provides a film of air which flows over the outer surface of the inner shroud, thereby cooling it.

Many modifications and variations of the present invention are possible in light of the above techniques. It is therefore to be understood that within the scope of the appended claims, the invention may be practiced otherwise than as specifically described.

I claim as my invention:

1. A gas turbine of the type having a turbine cylinder containing a plurality of stationary vanes and rotating

blades, said vanes and blades defining an annular flow path therebetween, said vanes circumferentially disposed in a row surrounding a rotating shaft and extending into said annular flow path;

each of said vanes having a radially inboard end, there being an inner shroud at each of said radially inboard ends;

each of said inner shrouds having first and second approximately axially oriented edges, said first and second edges of each pair of adjacent inner shrouds forming a circumferential gap, a slot being formed in each of said first and second edges;

each of said inner shrouds having inner and outer surfaces, said inner surfaces of said inner shrouds forming a shroud cavity;

a supply of high pressure air to said shroud cavity;

means for regulating the leakage of said high pressure air from said shroud cavity through each of said circumferential gaps between adjacent inner shrouds, characterized by:

a strip seal for each of said circumferential gaps, each of said strip seals having two longitudinal edges;

a sealing surface along each of said longitudinal edges, said sealing surfaces of each of said strip seals residing in said slots of two of said inner shrouds which are adjacent, one of said sealing surfaces residing in one of said slots and the other of said sealing surfaces residing in the other one of said slots whereby each of said strip seals spans one of said circumferential gaps; and

a plurality of intermittent reliefs in each of said sealing surfaces, the size and quantity of which being variable to obtain the leakage flow desired.

2. A gas turbine according to claim 1 wherein each of said strip seals comprises a dumbbell-shaped cross-section having cylindrical portions, each of said cylindrical portions extending the length of each of said seals, the diameter of said cylindrical portions being approximately that of the width of said slots, thereby forming said sealing surfaces.

3. A gas turbine having a turbine cylinder containing a plurality of stationary vanes and rotating blades, said vanes and blades defining an annular flow path therebetween, said vanes circumferentially disposed in a row surrounding a rotating shaft and extending into said annular flow path;

each of said vanes having a radially inboard end, there being an inner shroud at each of said radially inboard ends;

each of said inner shrouds having first and second approximately axially oriented edges, said first and second edges of each pair of adjacent inner shrouds forming a circumferential gap, a slot being formed in each of said first and second edges;

each of said inner shrouds having inner and outer surfaces, said inner surfaces of said inner shrouds forming a shroud cavity;

a supply of high pressure air to said shroud cavity;

a radial barrier extending circumferentially around said shroud cavity and extending into said shroud cavity, said radial barrier restricting the flow of said high pressure air supplied to said shroud cavity from flowing downstream past said barrier, said radial barrier having front and rear faces, a portion of each of said circumferential gaps being downstream of said radial barrier;

means for distributing said high pressure air to said portion of each of said gaps downstream of said radial barrier, comprising:

means for regulating the leakage of said high pressure air from said shroud cavity through each of said circumferential gaps, said regulating means disposed in each of said circumferential gaps and retained in said slots in said first and second axially oriented edges of said inner shrouds;

a plurality of holes in each of said inner shrouds, a portion of said holes in each inner shroud extending from said inner surface to said slot in said first approximately axially oriented edge and remaining portion of said holes extending from said inner surface to said slot in said second approximately axially oriented edge;

a plurality of holes in said radial barrier, extending from said front to said rear face of said barrier; and

a manifold for each of said inner shrouds, each of said manifolds connecting each of said holes in said radial barrier to said holes in its respective inner shroud.

4. A gas turbine according to claim 3 wherein the size of said holes in said radial barrier are variable to obtain the leakage flow desired.

5. A gas turbine according to claim 3 wherein each of said manifolds comprises a containment cover, each of said containment covers affixed to said inner surface of its respective inner shroud.

6. A gas turbine according to claim 3 wherein said radial barrier is comprised of a plurality of support rails, one of said support rails emanating from said inner surface of each of said inner shrouds.

7. A gas turbine comprising:

a plurality of vanes, said vanes arranged in a circular pattern so that each of said vanes has two other of said vanes adjacent to it, each of said vanes having a radially inboard end;

an inner shroud at said radially inboard end of each of said vanes, each of said inner shrouds having two approximately axially oriented edges, said approximately axially oriented edges of each pair of adjacent inner shrouds forming a circumferential gap, each of said shrouds having first and second portions;

a high pressure air supply, said high pressure air supplied to said first portion of each of said inner shrouds, said second portion of each of said inner shrouds not supplied with said high pressure air;

a plurality of slots, one of each of said slots disposed in each of said approximately axially oriented edges of said inner shrouds;

a strip seal for each of said circumferential gaps, each of said strip seals having two longitudinal edges, each of said edges forming a sealing surface, each of said strip seal disposed in its respective circumferential gap, each of said sealing surfaces being retained in said slots, whereby each of said strip seals spans its respective circumferential gap, a portion of each of said strip seals being located in said second portion of each inner shroud;

at least one relief in each of said sealing surfaces; and

a plurality of manifolds connecting said high pressure air to said portion of each of said strip seals located in said second portion of each inner shroud.

8. A gas turbine of the type having a turbine cylinder containing a plurality of stationary vanes and rotating blades, said vanes and blades forming an annular flow

path therebetween; a plurality of stationary members circumferentially arranged in a row surrounding a rotating shaft and forming a portion of said annular flow path, each of said stationary members being separated from each adjacent stationary member by a gap formed therebetween; and regulating means for regulating leakage through said gaps, said regulating means comprising:

a plurality of strip seals, each of said strip seals disposed in one of said gaps, each of said strip seals having first and second substantially longitudinal edges, a sealing surface along each of said longitudinal edges, each of said sealing surfaces having at least one relief, the size of said at least one relief being variable to obtain the degree of leakage desired, each of said sealing surfaces along said first longitudinal edges being in contact with one of said stationary members, each of said sealing surfaces along said second longitudinal edges being in contact with said adjacent stationary member forming said gap, whereby each of said strip seals spans one of said gaps.

9. A gas turbine according to claim 8 wherein said at least one relief comprises a plurality of intermittent reliefs in each of said sealing surfaces.

10. A gas turbine according to claim 8 further comprising first and second approximately axially extending edges formed in each of said stationary members, there being a slot in each of said axially extending edges, each of said longitudinal edges of said strip seals being disposed in one of said slots.

11. A gas turbine comprising a turbine cylinder containing an annular flow path, an annular cavity and a rotating shaft; a plurality of stationary members separating said annular flow path from said annular cavity, said stationary members circumferentially arrayed around said rotating shaft; each of said stationary members being separated from each adjacent stationary member by a circumferential gap; a radial barrier extending circumferentially around said annular cavity and dividing said annular cavity into first and second portions; first and second leakage paths between said second portion of said annular cavity and said annular flow path, said second leakage paths being formed by each of said circumferential gaps; means for regulating leakage of high pressure air through each of said second leakage paths, said regulating means comprising a seal with reliefs for leakage of air therethrough; a supply of high pressure air to said first portion of said annular cavity; and means for flow communication of said high pressure air between said first portion of said annular cavity and each of said second leakage paths, said flow communication means having means for preventing said high pressure air in said flow communication from communicating with said second portion of said annular cavity.

12. A gas turbine according to claim 11 wherein said stationary members comprise stationary vanes disposed in said annular flow path, each of said vanes having a radially inboard end, said stationary members forming an inner shroud at each of said radially inboard ends.

13. A gas turbine according to claim 11 further comprising a housing encasing said rotating shaft and forming a portion of said annular cavity, said radial barrier extending from each of said stationary members to said housing, thereby preventing flow of said high pressure air from said first to said second portions of said annular cavity.

14. A gas turbine according to claim 13 wherein said means for flow communication comprises a plurality of holes in said radial barrier and a manifold for each of said stationary members, each of said manifolds being in flow communication with one of said holes and one of said second leakage paths.

* * * * *